THE DEVELOPMENT OF AN APPARATUS TO INVESTIGATE THE EFFECT OF THE VARIATION OF FLOW ANGLE ON CONVECTION HEAT TRANSFER AND FRICTIONAL RESISTANCE OF A BOILER CONVECTION TUBE BANK.

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A	free flow area of tube bank, sq. ft.
a	exponent of Reynolds number
b	exponent of Prandtl number
C_p	specific heat, BTU/lb. - ^oF.
D	diameter, ft.
G	mass flow, lb./sq. ft. - hr.
h	heat transfer coefficient, BTU/sq. ft. - hr. - ^oF.
H_1	pressure preceding thin plate orifice, in. water
ΔH_f	pressure drop through thin plate orifice, in. water
k	thermal conductivity, BTU/ft. - hr. - ^oF.
K_1	flow coefficient
LMTD	log mean temperature difference
m	subscript denoting mean value
ΔP	pressure drop through tube bank, in. water
P	pressure, lb./sq. ft.
p_b	barometric pressure, lb./sq. ft.
Q	heat transferred, BTU/hr.
S	effective heat transfer surface, sq. ft.
T	temperature, ^oF.
T_{av}	average temperature
T_f	film temperature
T_t	tube temperature

T_1	temperature of air entering test section
T_2	temperature of air leaving test section
U_c	coefficient of convection heat transfer, BTU/sq. ft. - hr. - ^{o}F.
V_a	velocity of air, ft./sec.
W_a	weight rate of flow of air, lb/hr.
W_w	weight rate of flow of water, lb./hr.
Y_1	expansion factor
α	constant for heat transfer equation; also the angle of inclination of the tube bank
β	ratio of orifice diameter to pipe diameter
ρ	density of medium, lb./cu. ft.
μ	dynamic viscosity, lb./hr. - ft.
N_{NU}	nusselt number $= \frac{hD}{k}$
N_{RE}	Reynolds number $= \frac{GD}{\mu}$
N_{PR}	Prandtl number $= \frac{C_p \mu}{k}$

The work done by Huge and Pierson, upon which equation (2) is based, was accomplished with tube banks angled 90 degrees to the direction of flow of the cooling medium. No investigation has been made to date to determine the effect of changing the angle of the convection tube bank relative to the direction of gas flow. Thus the purpose for the development of the present apparatus is to aid in the evaluation of the heat transfer rate as affected by inclination to flow. The proposal is to introduce some factor, F_α , the angle correction factor, where:

$$F_\alpha = \frac{\alpha_c \text{ (for angle } \alpha \text{)}}{\alpha_c \text{ (for cross flow)}}$$

F_α is to be applied to equation (2) as an additional term.

Pierson obtained frictional resistance data for various tube arrays, all of which were angled at 90 degrees to the fluid flow. He expressed the results in terms of friction factor, f, where:

$$f = \frac{10.84 \times 10^8 \rho \Delta P}{N G^2} \qquad \text{(A)}$$

Concurrently with the investigation of heat transfer variation with angle of inclination, the variation of frictional resistance will be determined.

THEORY

The problem of heat transfer by a fluid passing through the convection tube bank is one of forced convection. For forced convection heat transfer, the heat is conducted through a film, so the thermal conductivity, k, of the fluid will be a factor. The film thickness depends on the mass velocity, G, through the section, the tube diameter, D_t, and the viscosity, μ, of the fluid. For a given quantity of heat transferred, the temperature rise of the fluid depends on the specific heat at constant pressure, C_p, of the fluids. If ϕ represents a function, then the following expression for the coefficient of heat transfer may be written:

$$h = \phi\left(G, D_t, \mu, c_p, k, \ldots\right) \tag{3}$$

This expression may be correlated through methods of dimensional analysis.

For conditions of forced convection it has been shown that satisfactory correlation of data may be obtained by an equation of the Nusselt type:

$$\frac{hD}{k} = \phi\left(\frac{GD}{\mu}, \frac{c_p \mu}{k}, \ldots\right) \tag{4}$$

Expressed in terms of power functions, equation (4) becomes:

$$\frac{hD}{k} = \alpha \left(\frac{GD}{\mu}\right)^{a} \left(\frac{c_p \mu}{k}\right)^{b}$$

which is equation (1). In familiar terms, equation (1) is:

$$N_{NU} = \alpha \left(N_{RE}\right)^{M} \left(N_{PR}\right)^{N} \tag{5}$$

As stated in the Introduction, equation (5) has been shown to be:

$$N_{NU} = 0.292\ F_a\ F_d\ (N_{RE})^{.6}\ (N_{PR})^{1/3} \tag{2}$$

In equation (2), F_a, the arrangement factor, is a function of tube arrangement (in line or staggered), longitudinal and transverse pitch, and Reynolds number. F_d, the depth correction factor, is a function of the number of tube rows traversed by the fluid and equals 1.0 for ten or more tube rows.

Since equation (2) is notably empirical in its approach to the convection heat transfer problem, no attempt will be made to make a theoretical or analytical prediction of the effect upon it of variation of the flow angle. Instead, the approach will be confined to the same experimental methods by which equation (2) was originally developed.

The use of equation (A) to evaluate friction factor follows the practice of Pierson, Huge, and Grimison. Equation (A) is a variation of the Fanning equation:

$$f = c\left(\frac{G\ D}{\mu}\right)^{x} \tag{B}$$

According to equation (B), when friction factor is plotted on a logarithmic scale as a function of Reynolds number, the mean values should fall on a straight line. Reference to plots of this type given by Pierson, Huge, and Grimison indicates that the friction factor does not plot as a straight line; rather the plot tends to "droop" for high values of Reynolds

number. Consequently, equation (A) has been adopted as a conventional method for evaluating the effect of frictional resistance.

It is apparent that the temperature at which f is evaluated has a large effect on its value. Gas temperature has direct effects on the values of both gas density and viscosity which in turn directly affect evaluation of friction factor and Reynolds number respectively. Evidently all three of the above investigators experienced difficulty in correlating values of f, since all three concur in a "conventional" film temperature at which to evaluate gas density and viscosity. This film temperature is given as tube temperature minus eight tenths of the mean temperature difference between tubes and gas for staggered arrangements and as tube temperature minus nine tenths of the mean temperature difference between tubes and gas for in-line arrangements. This convention seems somewhat unwieldly in its application, but Grimison states: "The validity of the general relation is clearly indicated by the comparisons of tests on large and small tubes ......" Be that as it may, Grimison's attempt at correlation of the results obtained by a number of investigators shows variations among them of up to thirty or forty percent at a given Reynolds number.

In the evaluation of frictional resistance as in the evaluation of heat transfer, the approach employed by previous investigators is predominantly empirical. From a purely pedantic viewpoint, such a procedure is indefensible; but as a practical matter, it has been shown that the results are reproducible and reasonably consistent, so that they may be used with some confidence in the design of heat transfer equipment. Little more can be

asked of any engineering data. Consequently the authors will use the same procedure in evaluating frictional resistance data as that outlined by Grimison.

METHOD OF APPROACH AND DESIGN OF APPARATUS

Basing the investigation on determining the effect upon equation (2) of variation of the angle of inclination, the problem then was to build an apparatus which would be:

1) Capable of erection and operation in the Haeberle Laboratory at the Webb Institute of Naval Architecture.
2) Capable of close correlation with the modified Grimison equation, (eq. 2), for a 90 degree flow angle.
3) Capable of rotation of the tube bank with respect to fluid flow to determine an inclination factor.
4) Capable of attaining reasonably high Reynolds numbers, preferably in excess of 10,000.

The present investigation was begun by Mulford and Graap (Webb '61) (7). The apparatus which was built to facilitate the investigation is described in Appendix (G). The cooling medium to be used was water, which, it was felt, would avoid the effects of the sensitivity of air to the ambient atmospheric conditions. Steam was used as a high temperature source due to its ready availability in the laboratory.

The design of the equipment was based upon an assumption that the film conductance of condensing steam was high enough to maintain the external tube temperature constant at approximately 212 degrees F. This assumption

proved to be unfounded in fact. The temperature of the tube external surface, as measured by various thermocouple configurations, was found to vary with both Reynolds number and water temperature, approaching the temperature of the flowing water. Since the work done by Pierson and Huge was carried out with a finite temperature difference between the flowing fluid and the tube wall, the conditions were different from those established by the Mulford-Graap apparatus. Thus the applicability of the apparatus to an investigation based on the Grimison equation was doubtful; indeed, correlation of data obtained from the Mulford-Graap apparatus with the Grimison equation could not be obtained, even after a lengthy series of alterations.

Since the film conductance of air is much less than that of water or wet steam, it was thought likely that if air were used instead of water as the cooling medium, the tube temperatures would in fact approach the temperature of condensing steam. Indeed, the previous investigators used air in their experiments, and air had been used in past experiments at Webb Institute with good results. An aircraft turbo-supercharger, converted for use with steam, was available in the Haeberle Laboratory as a source of air supply.

The blower was connected to the inlet of the Mulford-Graap apparatus, and an experimental run was made. The test was encouraging, since the results were in excellent agreement with the Grimison equation. In addition, the temperature rise across the test section was found to be large, about 60 degrees, so that the problem of instrumentation was minimized. Unfortu-

nately, the apparatus as constructed was not readily adaptable to the use of air. The small size 3" orifice meter and the long run of small supply line conducting air to the apparatus caused excessive losses of air pressure, so that values of Reynolds number obtainable were at the extreme lower end of the desired range. Hence it was decided to build a new apparatus.

The authors had previously designed a new tube bank for the existing apparatus in order to overcome a number of shortcomings of the original bank. This new tube bank (Appendix A) had been already constructed for the authors by Mr. Duncan Robb in the Webb Institute machine shop. It was available and ready for use. Consideration of available time made it desirable to investigate the feasibility of using this ready made tube bank. The tube bank was ten rows deep so that the depth correction factor (eq. 2) would equal 1.0. The new tube bank had been designed for a test section of inside dimensions 8 3/8" square, so that the size of test section was tentatively determined.

The adequacy of the air supply was then considered. It was proposed to employ an orifice meter made of standard 6" pipe. This was the largest size which could conveniently be handled with available facilities. A threaded flange was already available for mounting the orifice plate, and further it was desired to use the largest possible size of pipe in order to reduce line losses to a minimum. It was also desired to make the

orifice meter as long as possible to promote accuracy in measurement. The physical dimensions of the apparatus were controlled by the available space in the laboratory. It was decided to make the apparatus in the shape of an L with an elbow between the orifice meter and the test section; this arrangement was dictated primarily by the available space but was convenient in that it was possible to shield the upstream thermometer from direct radiation from the tube tank. Requirements for access and movement of other equipment in the laboratory made it necessary to provide eight feet of clear head room across the center aisle and to provide a section of ducting which could be removed if necessary; it was decided to use six feet of six inch stove pipe for this purpose. All of these considerations led to the dimensions given in the Description of Apparatus, Appendix A. The test section proper was eight feet long with stove pipe elbows at each end. The orifice meter was twelve feet long. The removable section consisted of four two foot sections of six inch stove pipe with one elbow.

From a thesis written by Gorman and Peterson (9), data on the capacity of the blower was obtained. The head losses on the proposed apparatus were estimated using the conventional hydraulic friction factors and the orifice meter calibration curve, Fig. 11, which was constructed for the purpose as described in Appendix D. An approximate system line was drawn for the apparatus. From this meager data, an estimate of maximum obtainable weight rate of flow of air of 4000 lbs per hour was made.

It then remained to determine whether the tentative design was capable of producing an acceptably high value of Reynolds number. From work done by Pierson and others, it was assumed that the tube surface temperature would be approximately that of the condensing steam. Measurements made while conducting the test on the original apparatus with air tended to confirm this assumption. Measured inlet and outlet air temperatures from the same run were used to give a realistic estimate of the temperature range to be expected.

Using the above assumptions, the maximum Reynolds number that could be expected was calculated as follows:

$$N_{RE} = \frac{G\,D}{\mu}$$

$$G = \frac{W}{A} \quad \text{lbs}/\text{ft}^2\text{-hr.}$$

In the direction of flow, the free flow area, A:

$$A = \frac{(8.375)\left(8.375 - \frac{22}{4}\right)}{144} = 0.1672\ \text{ft}^2$$

$$W = W_{max} = 4000\ \text{lbs/hr.}$$

$$G = \frac{4000}{.1672} = 23900\ \text{lbs/ft}^2\ \text{hr.}$$

Measured temperature values from the test run:

$T_1 = 90^\circ$ F

$T_2 = 150^\circ$ F

$T_t = 212^\circ$ F

Assume specific humidity = 50 grains/lb. dry air.

$$\Delta T = T_2 - T_1 = 150 - 90 = 60^\circ F.$$

$$T_{AV} = \frac{T_1 + T_2}{2} = \frac{150 + 90}{2} = 120^\circ F.$$

for $T_{AV} = 120^\circ F$ and sp. humidity = 50 :

$$c_p = 0.2416 \ BTU/lb.-^\circ F. \qquad (FIG. 13)$$

$$T_f = \frac{T_t + T_{AV}}{2} = \frac{212 + 120}{2} = 166^\circ F.$$

for $T_f = 166^\circ F.$:

$$k = 0.0174 \ BTU/FT-HR-^\circ F.$$

(FIG. 12)

$$\mu = 0.050 \ LB/HR-FT$$

$$N_{RE} = \frac{GD}{\mu} = \frac{(23900)(\frac{1}{48})}{.050} = \underline{\underline{9950}}$$

This maximum predicted Reynolds number is satisfactorily high although not as high as one might wish.

The steam system posed the remaining problem in the design process. The $1\frac{1}{2}$" supply line in the laboratory was easily extended to the site of the apparatus. The quantity required was calculated as follows:

$$Q = W_A C_p \Delta T$$

$$= (4000)(0.2416)(60) = 58000 \ ^{BTU}/_{HR}$$

$$W_{steam} = \frac{Q}{\Delta h_{fg}} = \frac{58000}{970.3} = 59.7 \ ^{LB}/_{HR.}$$

Thus the required flow of steam was about six pounds per header per hour. 3/8" copper tubing was considered entirely adequate for supply and drain lines to and from each header.

Working with the original apparatus, the authors had had experience with draining ~~and~~ the exhaust steam to the laboratory atmosphere. It was decided, therefore, to return the exhaust steam to the surface condenser located in the laboratory. This exhaust system was constructed by running a 1" copper line to the installed exhaust system in the laboratory.

ANALYSIS OF TEST RESULTS

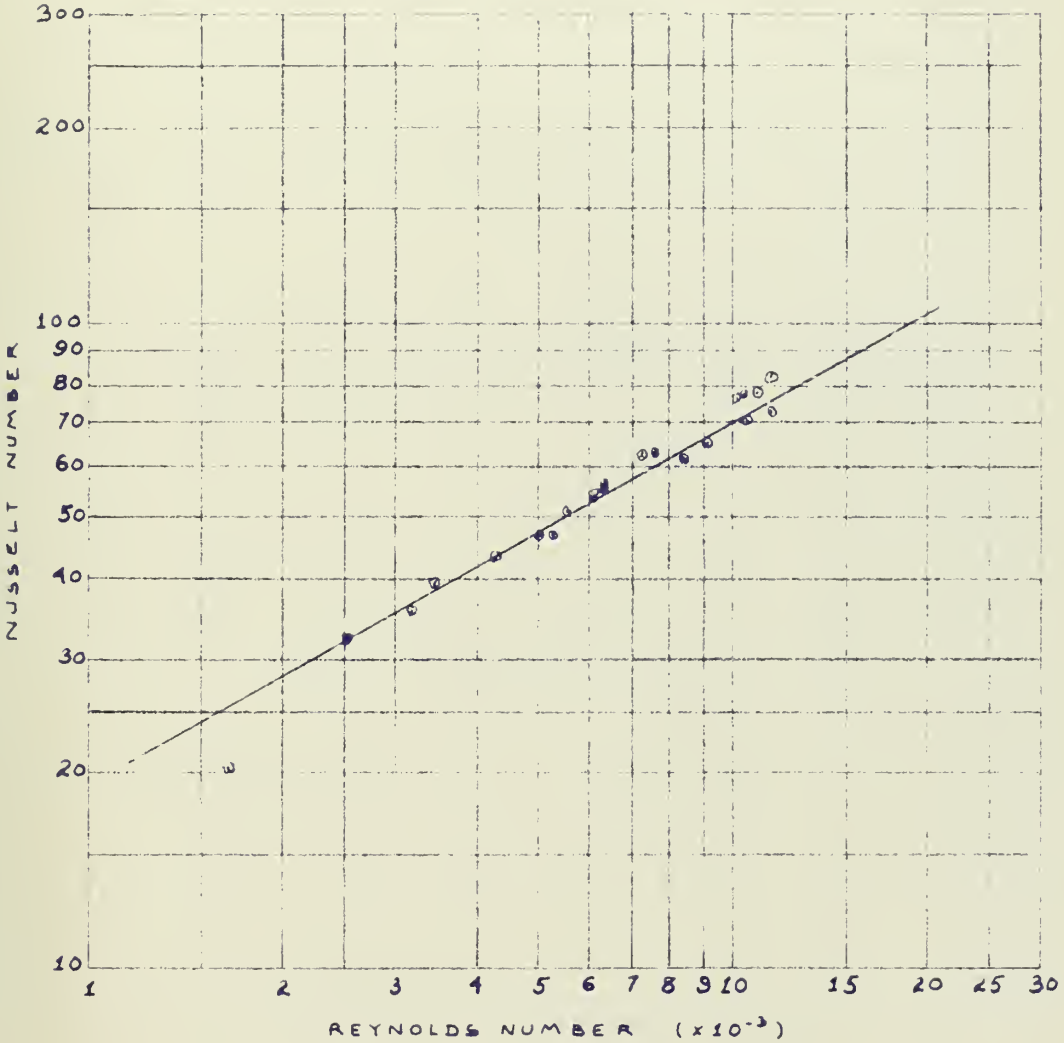
The variation of Nusselt number with Reynolds number, as determined by experiment, is shown as Figure 1. The solid line indicated on the plot is the theoretical variation of Nusselt number with Reynolds number as calculated from the modified Grimison equation (2). It will be noted that the theoretical line falls very close to the mean of the experimental points indicated; indeed, except for an obviously inconsistent point, all experimental data falls within ten percent of the Nusselt number predicted by the Grimison equation, and most of the points are closer. The inconsistent point, incidentally, fell off the mean line of the running plot kept during the course of taking data. It is included in the results primarily to indicate the value of the running plot in checking on the consistency of experimental data.

The experimental data includes points taken using both the 3" and 4" orifice plates in the flow meter. The fact that all points fall on the same mean line indicates that at least both orifice plates produce mutually consistent data. No experimental data was taken using the 2" orifice plate, because its range of operation is below the Reynolds number range of primary interest.

Draft loss through the tube bank was observed as part of the experimental data. This data is presented in two forms. Draft loss was converted to "friction factor" for Figure 2 and is compared to values given by Pierson and Grimison. It will be noted that values obtained are some

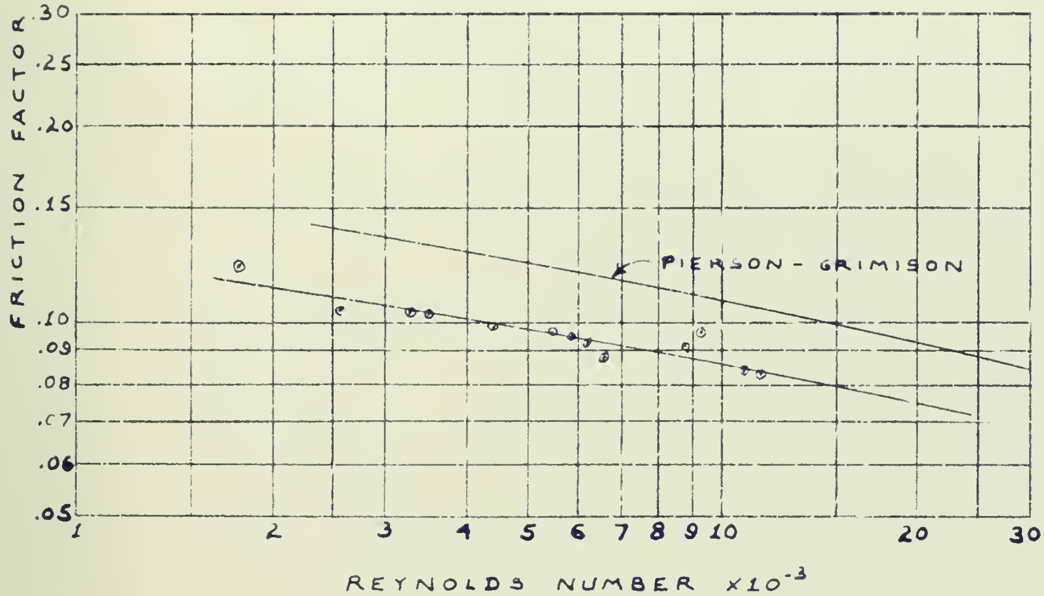
twenty percent lower than those obtained by Pierson and Grimison, although the curves are both of the same general shape. It should be noted that the values of Reynolds number indicated are adjusted to a lower film temperature (tube temperature minus eight tenths of the mean difference between tube and gas temperatures), as Pierson used. Pierson explained the use of this adjusted film temperature by noting that it lent a greater consistency to the experimental data.

Additionally, the draft loss data is presented in Figure 3 as a plot of draft loss in inches of water vs. Reynolds number (in this case, the unadjusted Reynolds number). It is felt that this "raw" data may be of some use in predicting the characteristics of the present apparatus at higher Reynolds numbers, should a blower of higher capacity be made available for experimental use.



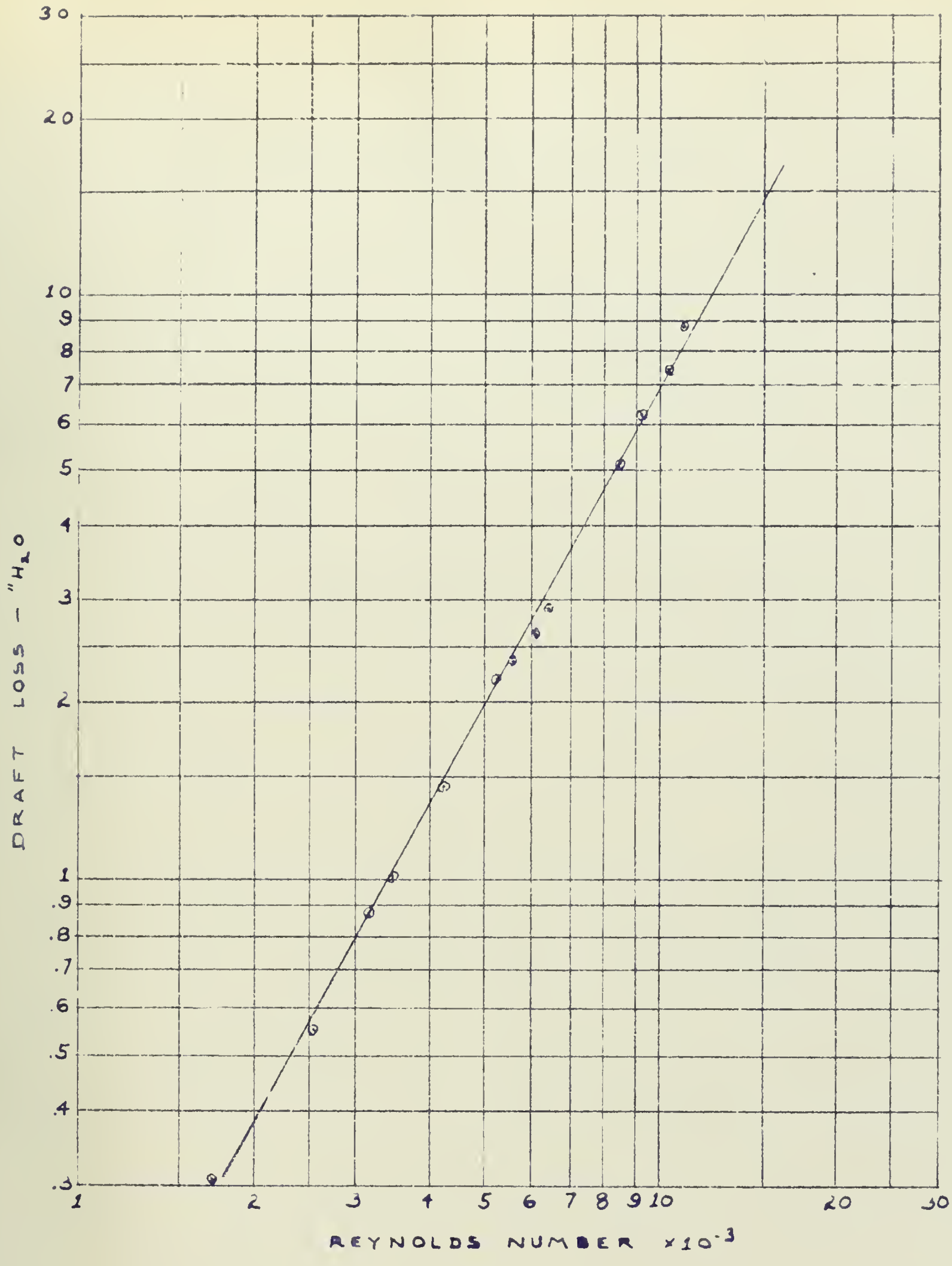
EXPERIMENTAL PLOT ~ N_{NU} VS N_{RE}
FLOW ANGLE = 90°

FIG. 1



EXPERIMENTAL PLOT OF FRICTION
FACTOR COMPARED WITH
PIERSON-GRIMISON DATA ~ $S_T/D = S_L/D = 1.5$

FIG. 2



EXPERIMENTAL PLOT OF
TUBE BANK DRAFT LOSS
90° FLOW ANGLE

FIG 3

CONCLUSIONS AND RECOMMENDATIONS

In the design and construction of this apparatus it was desired initially to obtain data for the ninety degree flow angle which could be correlated closely with the results obtained by previous investigators, all of whom investigated the process of convection heat transfer at the ninety degree angle. This step was necessary before any valid investigation of the effect of variation of flow angle could be undertaken.

In the case of the convection heat transfer coefficient (Nusselt number), this primary objective has been accomplished. Within bounds of reasonable experimental "scatter," the measured values of Nusselt number agree very closely indeed with values calculated from the modified Grimison equation which represents mean values obtained from previous work.

In the case of the frictional resistance, agreement with the previous data is somewhat less than striking. The experimental curve obtained exhibits the same characteristic shape and trend of the Pierson data but is some twenty percent lower, although the values of friction factor are of the same order of magnitude. The discrepancy may be due to some difference in the instrumentation, however Pierson gives little information as to precisely how he measured the tube bank pressure drop. We may infer, however, that Pierson had some difficulty in correlating his own data since it was necessary for him to make an arbitrary adjustment in his assumption of film temperature at which to evaluate his data. Moreover,

he states that he made <u>different</u> arbitrary adjustments for different tube arrangements. In view of these factors, it seems reasonable to expect some difficulty in precisely reproducing previous frictional resistance results.

It has been demonstrated that the apparatus is fully capable of producing results of acceptable accuracy. Further, it is reasonably simple to operate and is as economical to run as possible. The working fluid (air) is free, and steam is readily available in the laboratory at nominal cost. The authors have obtained a full set of data in a single afternoon, so that the actual running and data taking is the least time-consuming of the operations involved. The test section was designed with a view to easily varying the tube bank angle, so that the primary purpose of constructing the apparatus could be served with a minimum of effort. It is necessary, however, to remove the tube bank from the test section in order to remove and replace the seals at the tube bank ends.

Perhaps the versatility of the apparatus is its most useful feature. Since each row of tubes is separately mounted in separate headers, the arrangement as well as the flow angle can be varied. Thus the effects of varied back pitch and tube bank depth can be studied with both staggered and in-line arrangements. By making these adjustments, most of the experimental work which has been done on the convection heat transfer problem can be repeated with a single tube bank, and a high degree of accuracy

can be expected. These experiments, of course, are only additional possibilities for future use. This paper deals primarily with the design of an apparatus to determine the effect of flow angle on convection heat transfer.

In the absence of previous work in the field, it is difficult to predict the nature and magnitude of the effects of variation of flow angle on the rate of convection heat transfer. The apparatus was designed to find the answer experimentally. It seems possible that small variations in the angle will have little effect on the heat transfer. It also seems probable that at some angle of inclination a variation in heat transfer will become apparent. It is therefore recommended that the angle of inclination be varied in relatively large increments of from fifteen to twenty degrees until a noticeable effect is discovered.

The providing of satisfactory seals where the tube bank passes through the test section is the chief problem remaining. The present seals are made of notched wood strips glued in place and have been entirely satisfactory. When the flow angle is varied, however, the seals must be installed at the same angle relative to the tubes. Wood strips may be unmanageable and difficult to place in position so that some other method may prove desirable. The tube bank is clamped together so that no great mechanical strength is required for the seals. Synthetic sponge rubber - in view of the high tube temperature, natural rubber is not recommended - or some sort of plastic material such as is used for sealing electrical

connections may prove satisfactory. Some thought might be given to plaster of Paris cast seals. Whatever type of seal is used, the pressure to be contained is not great. Proper alignment of the seals relative to the tube bank is the major problem.

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APPENDIX A

DESCRIPTION OF APPARATUS

The test apparatus proper consists of two parts, the test section and the orifice meter. An over all view of the apparatus is shown in Photo no. 1, and a schematic diagram and plan and elevation drawings are given in Figures 4, 5, and 6, respectively. There are additionally a number of machinery items installed in the laboratory which, while necessary to the operation of the apparatus, are of secondary interest and will not be described here. They include the Clayton steam generator, the cooling tower, and miscellaneous pumps necessary to the operation of the above equipment.

The test section is essentially a wood box of square cross section. The inside dimensions are 8 3/8" square, and the section is eight feet in length. It is constructed of 3/4" exterior grade plywood, fastened together with glue, nails, and screws. The box is slotted on three sides at midlength to accomodate the tube bank, which passes through it. The test section is attached at the inlet end to the orifice meter through an elbow. The outlet end is open to the atmosphere through a short section of stove pipe and another elbow. The elbows are provided to shield the thermometers which measure inlet and outlet air temperatures from direct thermal radiation from the tube bank. A door is provided in the

test section to allow access to and removal of the tube bank. The test section is fitted with a water manometer to allow the air pressure drop across the tube bank to be measured. The test section is finished with two coats of white shellac and is mounted on a stand supported by a laboratory table (Photo No. 2).

The tube bank consists of ten rows of $\frac{1}{4}$" copper tubes with 22 tubes per row. Each row of tubes, as shown in Figure 7, is fitted with an inlet and exhaust header of $1\frac{1}{2}$" copper pipe. The tubes, as fitted into the headers, have a transverse pitch of $1\frac{1}{2}$ diameters. The tubes are clamped together and fitted with glued wood seals in a staggered arrangement with a longitudinal pitch of $1\frac{1}{2}$ diameters. The tube bank passes through the test section in a vertical plane and is supported on a stand made of angle iron (Photo no. 3). The tube bank may be readily removed from the test section in order to change the end seals and remount the assembly at a different angle of inclination. Each tube bank header is fitted with a steam supply or exhaust line of 3/8" copper tubing. The steam supply lines are led from a steam separator. The exhaust lines lead to a common header and discharge line which takes the condensate to the condenser. Pressure gages are fitted on the steam separator and exhaust header. A chromel-alumel thermocouple is soldered to the surface of one tube at each end of each row of tubes, so placed as to be inside the test section. Two of the thermocouples are hooked up to a direct reading Thwing pyrometer to give the tube temperature.

The orifice meter consists of a twelve foot section of 6" pipe with a threaded flange and taps 1 pipe diameter upstream and $\frac{1}{2}$ diameter downstream. Three orifice plates are provided with concentric holes 2", 3", and 4" in diameter. The flange and orifice plates are drilled, and centering pins are provided so that the plates may be accurately centered in the flange. The orifice plates may be readily changed by supporting the 6" pipe on one side of the flange with the crane in the laboratory (Photo no. 4) and removing three of the six bolts in the flange. The orifice meter is equipped with a differential water manometer to indicate the pressure drop across the orifice and an open water manometer to indicate the pressure on the upstream side. The orifice meter is connected to the test section with a 6" stove pipe elbow and to the blower with four lengths and an elbow of 6" stove pipe. The sections of stove pipe are made up with sheet metal screws, and the seams are soldered, so that leakage has been reduced to a negligible amount.

The blower is worthy of brief mention. It is a converted aircraft turbo-supercharger operating on steam from the Clayton steam generator. Its demonstrated capacity, operating with the apparatus under actual test conditions, is a maximum of about 4600 pounds of air per hour. This mass flow is sufficient for a Reynolds number of about 11000. The blower has produced a weight flow of somewhat in excess of 5000 pounds per hour, but the necessity for providing heating and air ejector steam reduces the maximum capacity. Air mass flow is varied by throttling the blower rather

than the air, because it was desired to keep line losses to a minimum and because of the expense of a six inch valve. The blower has operated with satisfactory stability under these conditions, and fineness of control is adequate. The blower with the stove pipe duct to the flow meter is shown in Photo no. 5.

PHOTOGRAPHS

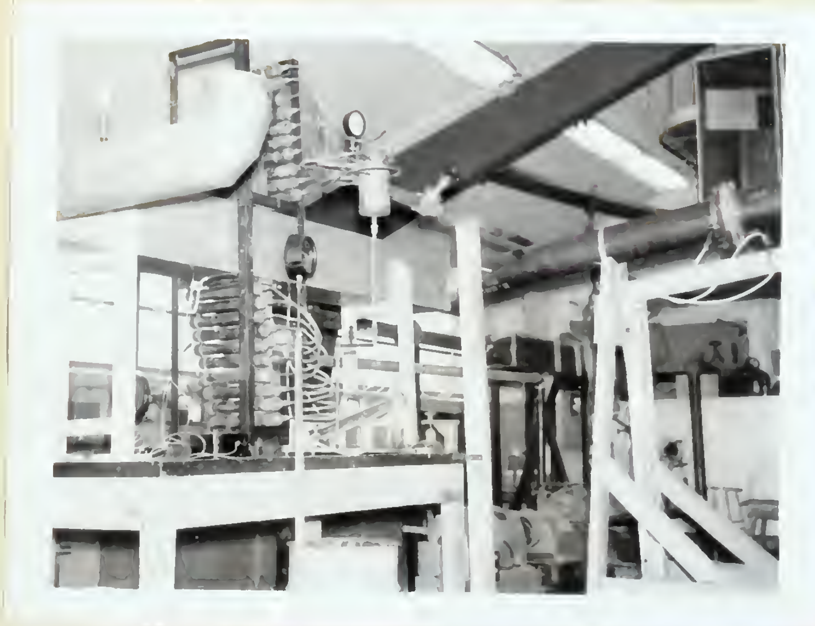


Photo no. 1 - Over all view of apparatus

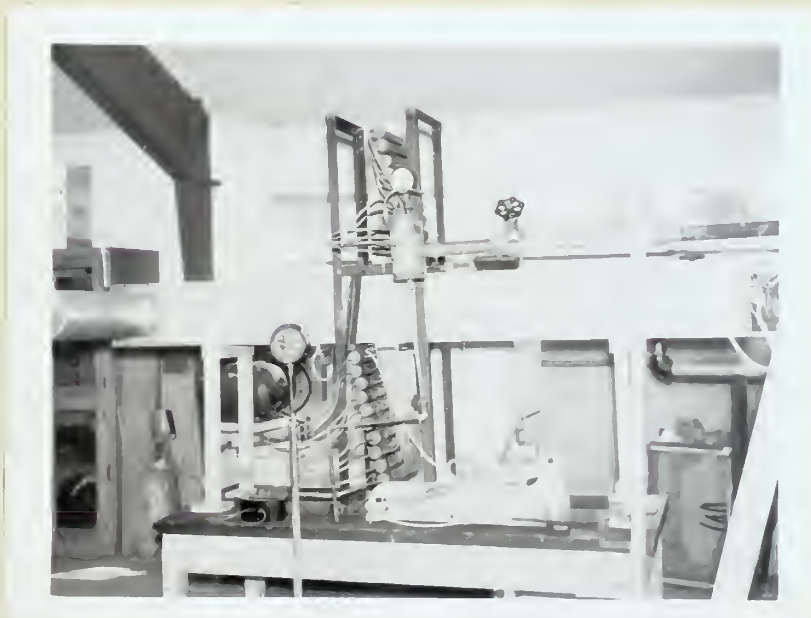


Photo no. 2 - Test section mounted on laboratory table

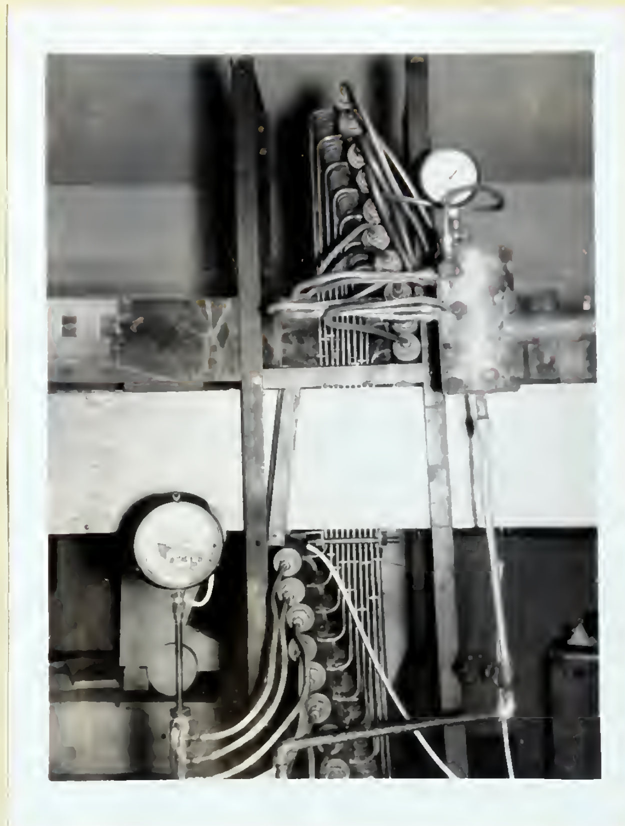


Photo no. 3
Closeup view of tube
bank

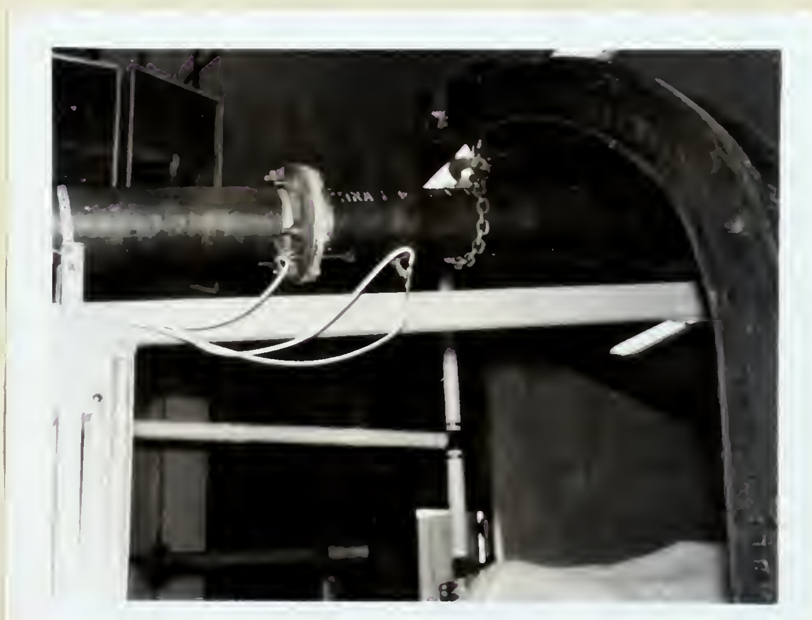
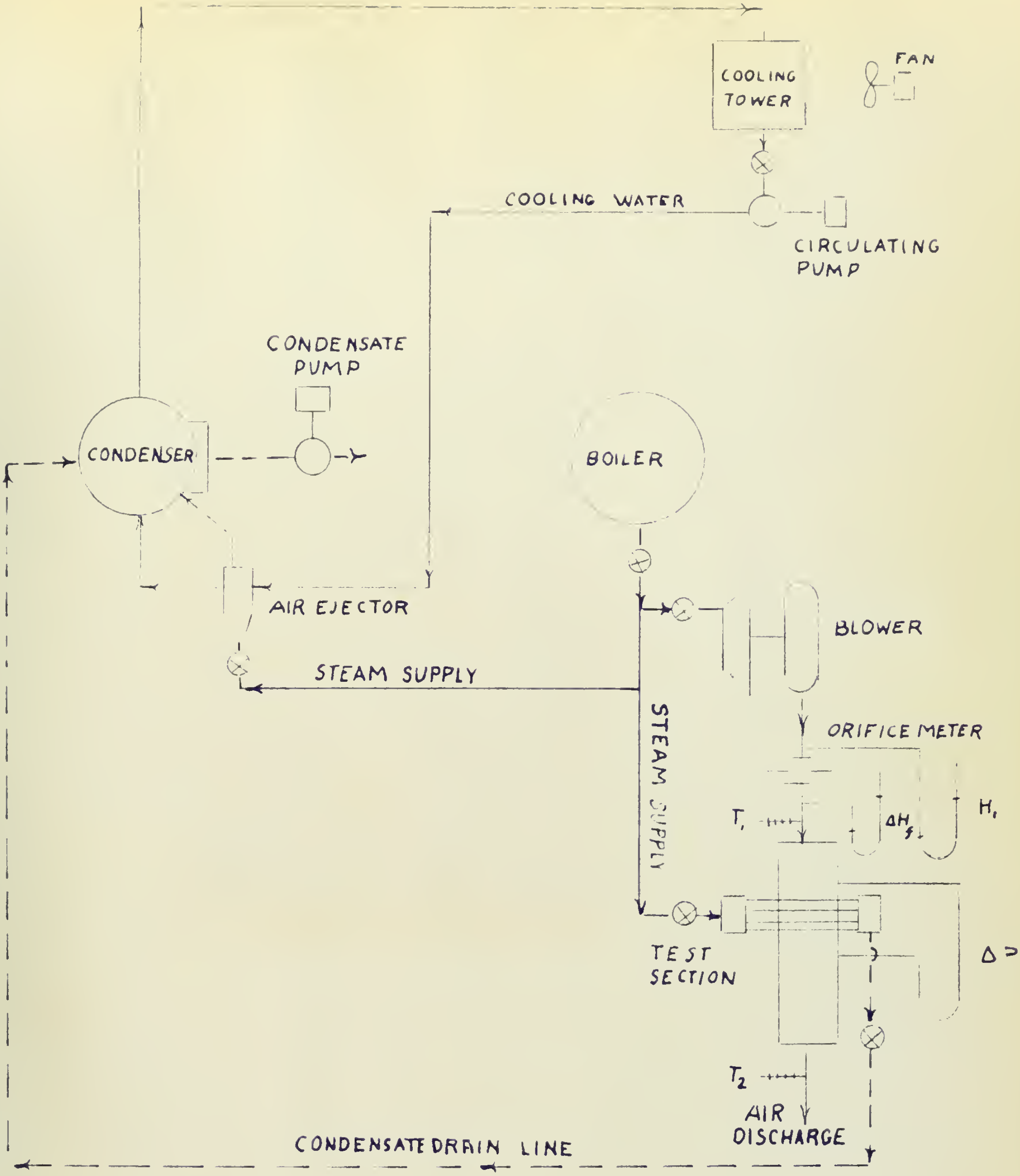


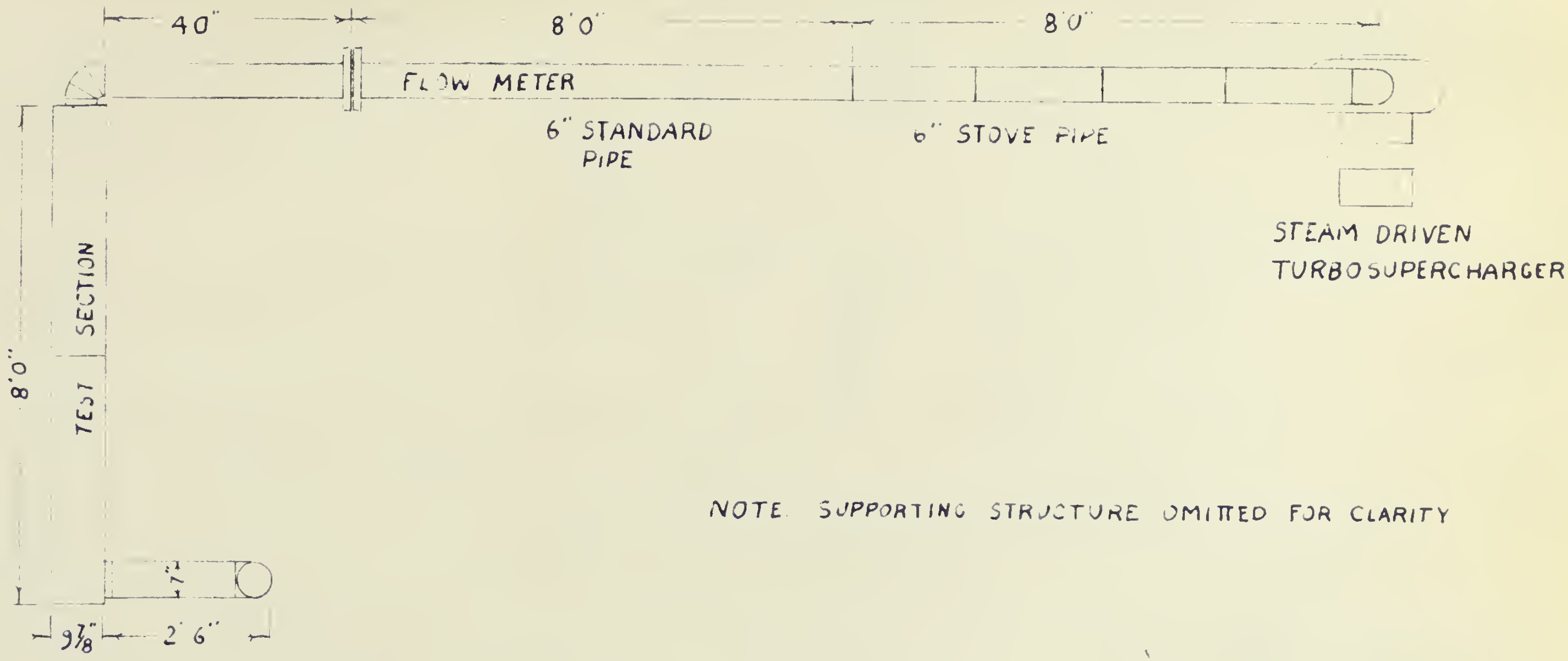
Photo no. 4 - Flow meter showing crane in position to
change orifice plate

Photo no. 5 - Air supply blower



SCHEMATIC DIAGRAM OF TEST APPARATUS

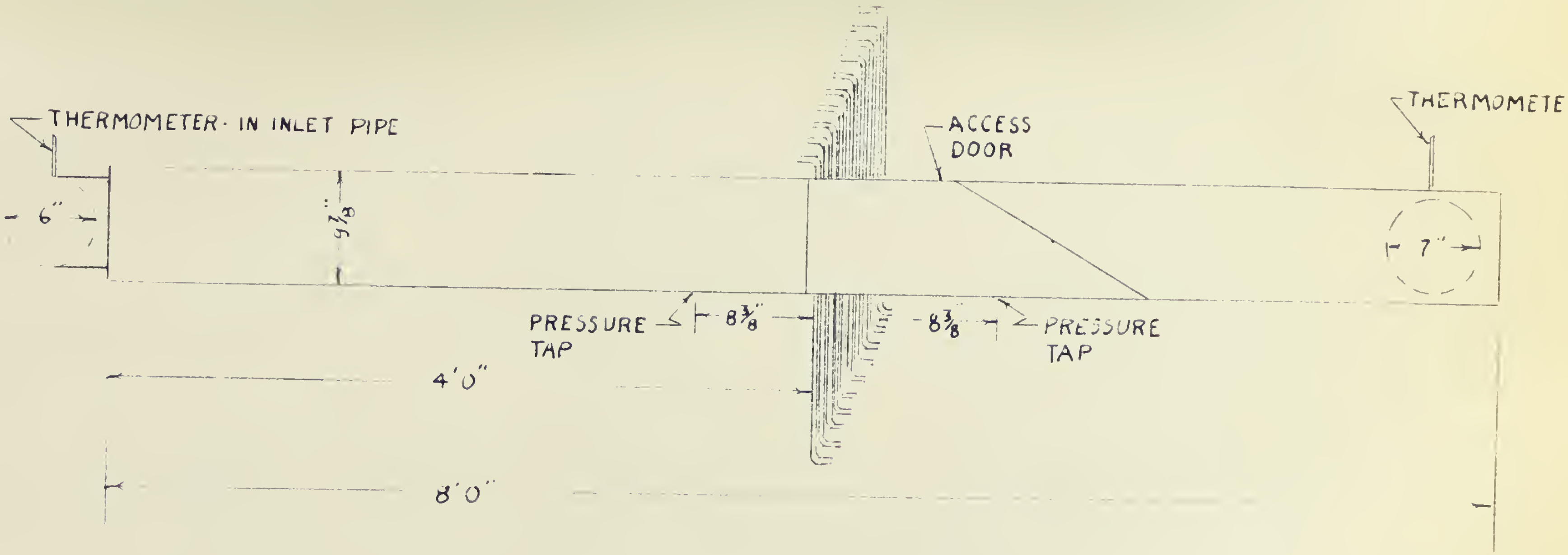
FIG. 4



NOTE: SUPPORTING STRUCTURE OMITTED FOR CLARITY

PLAN VIEW OF APPARATUS
SCALE: 1/3" = 1'

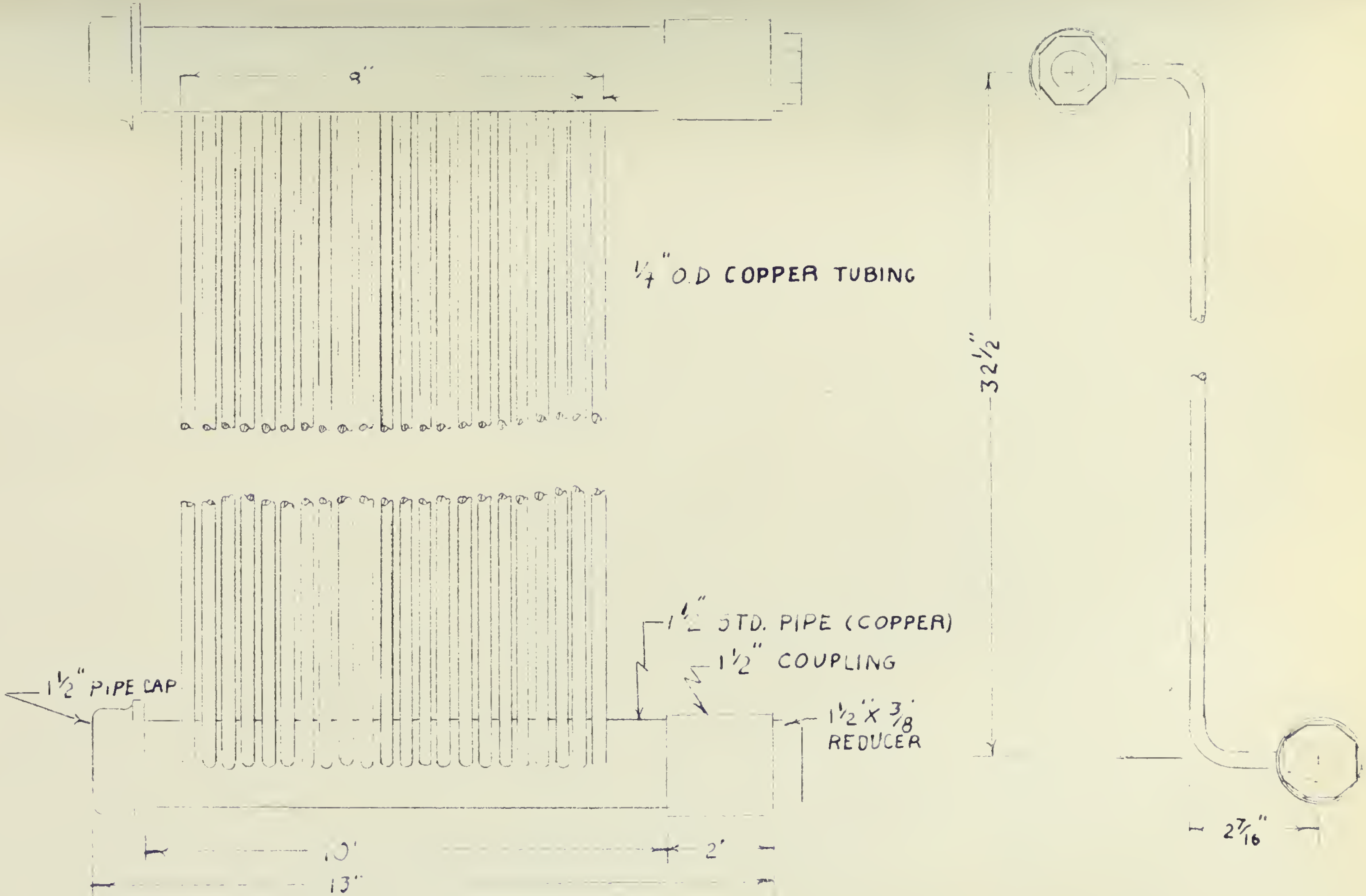
FIGURE 5



NOTE: TUBE BANK OF $\frac{1}{4}$" OD COPPER TUBES
HEADERS OF 1" COPPER PIPE
TRANSVERSE PITCH = $\frac{3}{4}$"
LONGITUDINAL PITCH = $\frac{3}{8}$"

ELEVATION OF TEST SECTION
SCALE: 1" = 1'

FIGURE 6



DETAIL DRAWING OF A TYPICAL TUBE BANK SECTION

SCALE: 1" = 3"

FIGURE 7

FIG. 7

APPENDIX A

CALIBRATION OF APPARATUS

All of the mercury-in-glass thermometers installed in the apparatus were checked for calibration in melting ice and boiling water. In addition, the direct reading Thwing pyrometer for indicating thermocouple temperatures was checked by the same method.

The orifice meter curves presented as Figs. 9, 10, and 11 were constructed from data contained in Reference (6). No attempt was made to calibrate the orifice meter since no practicable method of doing so was available. A standard orifice meter of a size to measure the weight rates of flow of primary interest was not available, so the meter constructed for the apparatus could not be checked by direct comparison. Since air was the working fluid, it was not feasible to weigh the discharge from the meter, as was done for the apparatus using water as the working fluid.

APPENDIX B

TEST APPARATUS OPERATING PROCEDURE

With some practice the apparatus can be placed in operation in something less than twenty minutes time. The individual operator may well wish to modify the schedule, but the following procedure is recommended:

1. Light off the Clayton steam generator in accordance with the instructions in the operating manual to be found in the laboratory. When the boiler pressure rises to about 140 psig, the main steam stop can be opened.
2. Open the turbo-blower throttle gradually until the blower throttle pressure is about 5 psig. This steam consumption stabilizes the operation of the boiler so that it will require little attention for the time being.
3. Start the reciprocating electric condensate pump.
4. Line up the condenser cooling water system, and start the circulating pump and cooling tower fans.
5. Line up and cut in one air ejector nozzle. One nozzle cut in slightly will produce a vacuum of about 12" which is entirely adequate to ensure proper drainage of the tube bank.
6. Drain the steam separator, and crack in the tube bank steam supply valve. It will be noted that the rapid drainage of condensate from the main steam line "drags" the boiler causing the

water level to surge out of sight. This condition is not dangerous, but it is recommended that the boiler be blown down to bring the water level back into the sight glass.

7. Check all systems for proper operation.

8. Increase blower speed to produce the desired amount of air flow. Note the boiler steam pressure carefully during this phase, and cut in the second burner and feed pump as necessary to maintain operating pressure. For low rates of air flow, it is not desirable to use the second burner; since the boiler will cut in and out automatically causing fluctuation in steam pressure.

9. When the desired air flow has been obtained, regulate the steam supply to the tube bank to obtain the proper tube temperature. Tube temperature is measured by thermocouples attached to the tube surfaces and may be read directly from the Thwing pyrometer.

10. When the inlet and outlet air temperatures have been stabilized data may be taken. It has been found that when data is taken initially at the highest attainable rate of air flow, the temperatures stabilize almost as quickly as the blower can be slowed down to the next desired point.

The securing procedure is the reverse of the above except that the blower may be left running at a slow speed until after the boiler has been secured in order to bleed off the steam pressure. Leave the main boiler stop cracked open during and after securing the boiler. Finally,

shut off the blower.

Testing procedure consists primarily of operating the apparatus at the desired condition and recording data. Inlet and outlet air temperatures are read from mercury-in-glass thermometers installed at the inlet and outlet, respectively, of the test section. Air pressure and pressure drops are read from the appropriate manometer.

The measurement of tube temperature is worthy of some discussion. It is obviously of paramount importance in determining the "driving temperature difference" between the tubes and the average air temperature. An error in tube temperature measurement causes an error of the same degree in the calculation of heat transfer coefficient. A survey of the literature on the subject and personal experience have convinced the authors that accurate measurement of the surface temperature of $\frac{1}{4}$" copper tubes is at least an extremely difficult problem. Pierson, in conducting the experiment upon which the current investigation is based, heated the tubes electrically and measured the temperature by change in resistance. This method, while not too sophisticated for the authors to cope with, was deemed excessively expensive in cost of electric power. Consequently it was decided to assume that the temperature of copper tubes containing condensing steam is equal to the condensation temperature of the steam. This assumption has proven to be reasonably valid in the case of the apparatus employing air as the heated fluid, but cautious operation is required to make the test conditions fit the assumption.

The installed thermocouples should be used to check the tube temperature. At present, two of the twenty thermocouples are hooked up, and it

is felt that they give a reasonably accurate average tube temperature. It is possible, with careful regulation of steam supply, to make them both indicate within about five degrees of 212° F. This condition normally obtains when the steam supply pressure is about 5 psig, and the outlet pressure gage indicates a slight positive pressure. For best results, it is recommended that condenser vacuum be maintained at 10-12 inches of mercury. A lesser vacuum does not drain the tube bank properly. When a vacuum higher than 15" Hg is maintained, it has been observed that no condensate emanates from the condensate pump discharge. The empirical conclusion follows that the condensate pump is incapable of draining the condenser when the vacuum is held higher than the recommended level. Tube temperature should be checked at each rate of air flow before data is taken and the steam supply adjusted as necessary.

At low rates of air flow it will be found that the boiler is not loaded heavily enough to maintain stable operation. The boiler cuts in and out automatically causing a fluctuation in steam pressure, and consequently in the air flow. It is therefore desirable to consume enough steam to keep the boiler in continuous operation. When data is to be taken at low rates of air flow, it is recommended that one or more of the many rotating and knee-action turbines located in the laboratory be idled to consume the excess steam.

It is suggested that during testing a rough plot be made of

$$\frac{T_t - T_1}{T_t - T_2}\sqrt{\Delta H_f} \quad vs. \quad \sqrt{\Delta H_f}$$

These terms are directly proportional to Nusselt number and Reynolds number respectively.

When plotted on log-log graph paper, the test points should form a straight line. This method will serve as a good check during operation on the consistency, if not the accuracy, of the experimental data, since:

$$W_A \sim \sqrt{\Delta H_f}$$

But $\quad Q = W_A C_p \Delta T \quad$ and $\quad u = \dfrac{Q}{S D}$

Where $\quad D = \dfrac{\Delta T}{\ln \dfrac{T_t - T_1}{T_t - T_2}}$

Therefore $\quad u = \dfrac{W_A C_p}{S} \ln \dfrac{T_t - T_1}{T_t - T_2}$

or since $\dfrac{C_p}{S}$ is nearly constant, $\quad u \sim \dfrac{T_t - T_1}{T_t - T_2} \sqrt{\Delta H_f}$

and since $\quad N_{NU} = \dfrac{u D}{k} \quad , \quad N_{NU} \sim \dfrac{T_t - T_1}{T_t - T_2} \sqrt{\Delta H_f}$

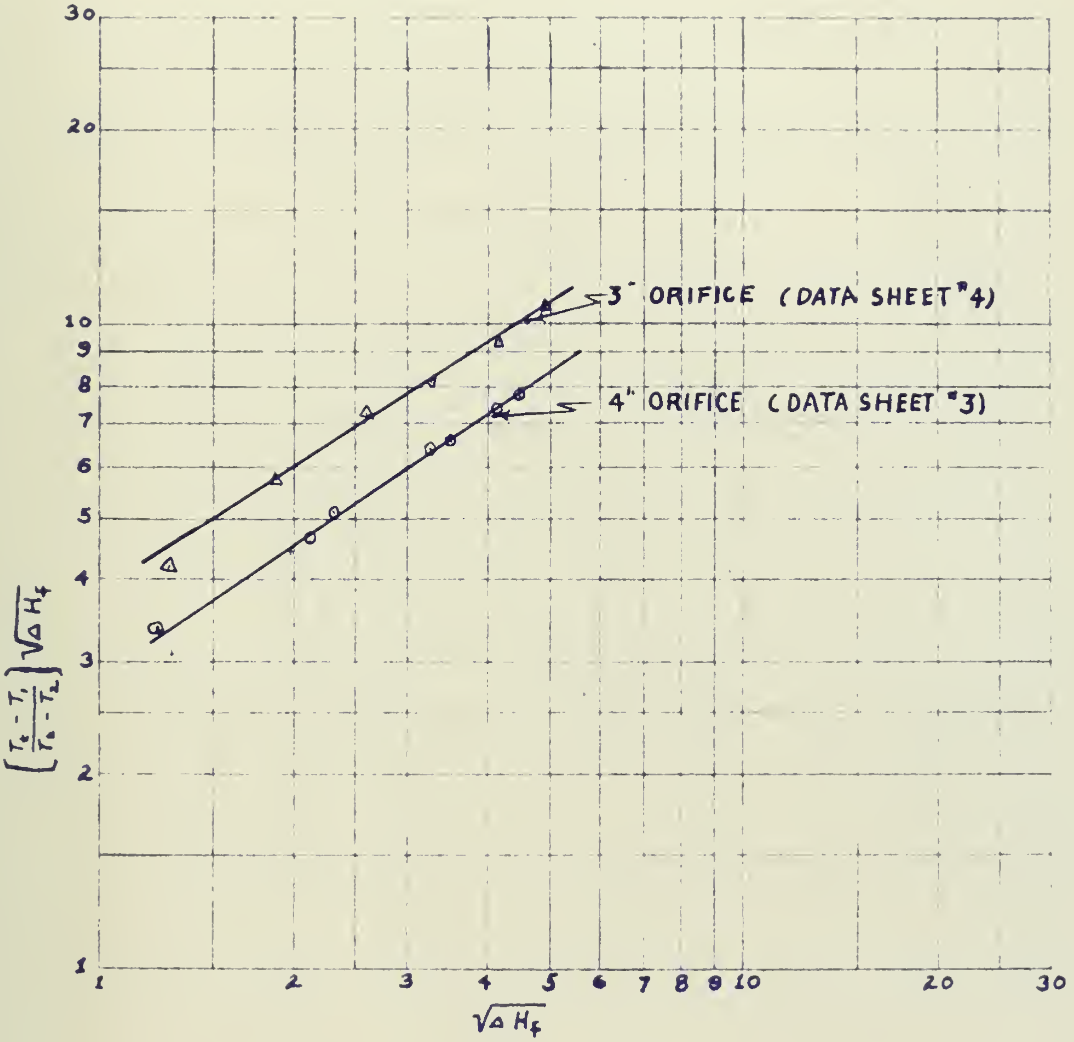
Also $\quad N_{RE} = \dfrac{G D}{\mu} \quad$ and $\quad G = \dfrac{W_A}{A}$

Therefore $\quad N_{RE} \sim \sqrt{\Delta H_f}$

A sample plot is included as Fig. 8. It will be noted that one point at the lower end of the range for the 3" orifice falls below the mean line of the other points. Data for this point was calculated and the results included in the Experimental Plot, Fig. 1, where the same point falls below the mean line at a Reynolds number of about 1700. This point is included in the data analysis for illustrative purposes.

It will be noticed that there are three different orifice plates pro-

vided for measuring air flow. Each orifice is applicable to a different range. Data should be taken with each orifice to the extreme ends of the range to obtain continuity. When sufficient data has been taken in one range, the blower may be secured and the orifice plate changed, after which testing can be quickly resumed. The authors have performed this operation twice in a two hour period and are confident that it will offer little difficulty.



NONDIMENSIONAL PLOT
HEAT TRANSFER VS.
AIR FLOW

FIG. 8

APPENDIX C

TEST DATA SHEETS

Appendix C

Test Data Sheet Number 1 Date: 1/31/62

Model tube bank at 90 degree flow angle

4" flow meter

Ambient Conditions

	Start	Finish	Average
Wet bulb temperature (°F.)	53.5	53.5	53.5
Dry bulb temperature (°F.)	60	60	60
Barometer ("Hg.)	30.08	30.08	30.08

Specific humidity 50 $\frac{\text{grains}}{\text{lb. dry air}}$

(1) Run No.	(2) ΔH_f "H_2O	(3) H_1 "H_2O	(4) ΔP "H_2O	(5) $\sqrt{\Delta H_f}$	(6) Temperature (°F.)				(7) $\frac{T_t - T_1}{T_t - T_2}$	(8) (5) × (7)
1	25.0	29.4	-	5.0	T_t T_1 Δ	204 104 100	T_t T_2 Δ	204 151 53	1.885	9.42
2	20.5	24.4	-	4.53	T_t T_1 Δ	208 91 117	T_t T_2 Δ	208 145 63	1.856	8.41
3	18.0	22.6	-	4.25	T_t T_1 Δ	210 89 121	T_t T_2 Δ	210 145 65	1.860	7.91
4	9.0	12.2	-	3.0	T_t T_1 Δ	210 76 134	T_t T_2 Δ	210 144 66	2.03	6.09

Appendix C

Test Data Sheet Number 2 Date: 3/14/62

Model tube bank at 90 degree flow angle

4" flow meter

Ambient Conditions

	Start	Finish	Average
Wet bulb temperature ($^oF.$)	60	59.5	59.75
Dry bulb temperature ($^oF.$)	66.5	65	65.75
Barometer ("Hg.)	29.75	29.75	29.75

Specific humidity 68 $\frac{\text{grains}}{\text{lb. dry air}}$

(1) Run No.	(2) ΔH_f "H_2O	(3) H_t "H_2O	(4) ΔP "H_2O	(5) $\sqrt{\Delta H_f}$	(6) Temperature ($^oF.$)		(7) $\frac{T_t - T_1}{T_t - T_2}$	(8) (5)×(7)
1	20.25	22	-	4.5	T_t 215 T_1 92 Δ 123	T_t 215 T_2 147 Δ 68	1.81	8.15
2	17.25	19.2	-	4.155	T_t 210 T_1 90 Δ 120	T_t 210 T_2 146 Δ 64	1.875	7.80
3	14.38	17.4	-	3.79	T_t 205 T_1 87 Δ 118	T_t 205 T_2 146 Δ 59	1.99	7.75
4	7.88	10.42	-	2.79	T_t 210 T_1 80 Δ 130	T_t 210 T_2 149 Δ 61	2.13	5.95
5	6.25	5.22	-	2.5	T_t 212 T_1 78 Δ 134	T_t 212 T_2 150 Δ 62	2.16	5.4
6	5.5	8.7	-	2.34	T_t 215 T_1 77 Δ 138	T_t 215 T_2 151 Δ 64	2.15	5.05
7	3.88	-	-	1.97	T_t 215 T_1 76 Δ 139	T_t 215 T_2 152 Δ 63	2.21	4.35

Appendix C

Test Data Sheet Number 3 Date: 3/23/62

Model tube bank at 90 degree flow angle

4" flow meter

Ambient Conditions

	Start	Finish	Average
Wet bulb temperature (^{o}F.)	5.95	64.5	62
Dry bulb temperature (^{o}F.)	67	71	69.0
Barometer ("Hg)	29.97	29.95	29.96

Specific humidity 71 $\frac{\text{grains}}{\text{lb. dry air}}$

(1) Run No.	(2) ΔH_f "H_2O	(3) H_2 "H_2O	(4) ΔP "H_2O	(5) $\sqrt{\Delta H_f}$	(6) Temperature (^{o}F.)	(7) $\frac{T_t - T_1}{T_t - T_2}$	(8) (5)×(1)
1	20-5/8	25.0	8-7/8	4.55	T_t 212, T_1 96.5, Δ 115.5; T_t 212, T_2 145, Δ 67	1.725	7.85
2	17-1/8	21	7-3/8	4.14	T_t 212, T_1 94, Δ 118; T_t 212, T_2 145.5, Δ 66.5	1.775	7.35
3	12-5/8	15.5	6-1/4	3.56	T_t 212, T_1 89.5, Δ 122.5; T_t 212, T_2 146.5, Δ 65.5	1.87	6.65
4	11-1/8	13.5	5-1/8	3.33	T_t 212, T_1 88, Δ 124; T_t 212, T_2 149, Δ 63	1.97	6.54
5	5.5	6-7/8	2.63	2.345	T_t 212, T_1 83, Δ 129; T_t 212, T_2 153, Δ 59	2.19	5.1
6	4.75	6-1/4	2.37	2.18	T_t 212, T_1 82.5, Δ 129.5; T_t 212, T_2 153, Δ 59	2.195	4.78
7	1-5/8	2-1/2	0.875	1.275	T_t 212, T_1 82, Δ 130; T_t 212, T_2 161, Δ 51	2.55	3.35

Appendix C

Test Data Sheet Number 4 Date: 3/23/62

Model tube bank at 90 degree flow angle

3" flow meter

Ambient Conditions

	Start	Finish	Average
Wet bulb temperature (°F.)	64.5	69.5	67
Dry bulb temperature (°F.)	71	78.5	74.75
Barometer ("Hg.)	29.95	29.95	29.95

Specific humidity 85 $\frac{\text{grains}}{\text{lb. dry air}}$

(1) Run No.	(2) ΔH_f "H_2O	(3) H_2 "H_2O	(4) ΔP "H_2O	(5) $\sqrt{\Delta H_f}$	(6) Temperature (°F.)		(7) $\frac{T_t - T_1}{T_t - T_2}$	(8) (5)x(7)
1	24-5/8	22.5	2.9	4.97	T_t 212 T_1 90.5 Δ 121.5	T_t 212 T_2 155.5 Δ 56.5	2.15	10.7
2	17	15-5/8	2.23	4.13	T_t 212 T_1 88 Δ 124	T_t 212 T_2 157 Δ 55	2.255	9.3
3	10-5/8	9-3/4	1.435	3.26	T_t 212 T_1 85.5 Δ 126.5	T_t 212 T_2 161 Δ 51	2.48	8.08
4	7-1/8	6-3/4	1.01	2.67	T_t 212 T_1 84 Δ 128	T_t 212 T_2 164.5 Δ 47.5	2.69	7.2
5	3-3/4	3-3/4	0.55	1.935	T_t 212 T_1 83 Δ 129	T_t 212 T_2 169 Δ 43	3.00	5.80
6	1-3/4	1-5/8	0.312	1.325	T_t 212 T_1 82 Δ 130	T_t 212 T_2 170.5 Δ 41.5	3.13	4.15

APPENDIX D

DERIVATION OF EQUATION FOR THE CALCULATION OF W_A FROM ΔH_f

As a basis for the development of a theoretical hydraulic equation, (6) the assumptions customarily made are: that the specific weight of an incompressible fluid is constant, and that the combined effects of resistance, non uniformity, and turbulence are negligible. Thus the initial equation is taken as

$$V_2^2 - V_1^2 = 2g(h_2 - h_1)$$

or if $h = h_2 - h_1$

$$V_2^2 - V_1^2 = 2gh \qquad (1)$$

Consider a liquid flowing through any two normal sections of a channel where the first section is considerably larger than the second. Then,

$$A_1 V_{M_1} = A_2 V_{M_2} = q_t \qquad (2)$$

or

$$V_{M_1} = q_t/A_1$$

and

$$V_{M_2} = q_t/A_2$$

where q_t (cfs) is the theoretical volume rate of flow.

From (1) and (2)

$$V_{M_2}^2 - V_{M_1}^2 = 2gh$$

$$V_{M_2}^2 = \frac{2gh}{1-(A_2/A_1)^2}$$

$$V_{M_2} = \sqrt{2gh}\left(\sqrt{\frac{1}{1-(A_2/A_1)^2}}\right) \qquad (3)$$

Also from equation (2)

$$q_t = A_2 \sqrt{2gh} \left(\frac{1}{\sqrt{1 - (A_2/A_1)^2}} \right) \tag{4}$$

If W_t (lb per sec) is the theoretical weight rate of flow the relation is:

$$W_t = \rho q_t$$

$$= A_2 \, \rho \sqrt{2gh} \left(\frac{1}{\sqrt{1 - (A_2/A_1)^2}} \right) \tag{5}$$

If a closed manometer was to be connected between sections 1 and 2 in the flow channel the difference in the level of the fluid in the manometer or the "differential" head is

$$h = \left(\frac{P_1}{\rho} + \Lambda_1 \right) - \left(\frac{P_2}{\rho} + \Lambda_2 \right) \tag{6}$$

where Λ_1 and Λ_2 are the heights of the gravity of A_1 and A_2 respectively above a base line. If the centers of gravity of A_1 and A_2 are at the same height above the base level, then $\Lambda_1 = \Lambda_2$ and equation (6) becomes

$$h = \frac{P_1 - P_2}{\rho} \tag{7}$$

substituting into equation (4)

$$q_t = A_2 \left(\sqrt{\frac{2g(P_1 - P_2)}{\rho}} \right) \left(\frac{1}{\sqrt{1 - (A_2/A_1)^2}} \right) \tag{8}$$

and

$$W_t = A_2 \left(\sqrt{2g\rho(P_1 - P_2)} \right) \left(\frac{1}{\sqrt{1 - (A_2/A_1)^2}} \right) \tag{9}$$

Equations (8) and (9) are the theoretical hydraulic equations for the rate of flow of an incompressible fluid through section A_2 in terms of the pressure difference $(P_1 - P_2)$ pounds per square foot, and the specific weight ρ pounds per cubic foot. The equations will now be modified to suit a flow meter.

A_1 and A_2 are circular with the diameters D_1 and D_2 respectively (in feet).

$$\frac{A_2}{A_1} = \left(\frac{D_2}{D_1}\right) = \beta^2 \tag{10}$$

and

$$\frac{1}{\sqrt{1-(A_2/A_1)^2}} = \frac{1}{\sqrt{(1-\beta^4)}}$$

$=$ velocity of approach factor

Let C = the discharge coefficient

$$= \frac{\text{actual weight rate of flow}}{\text{theoretical weight rate of flow}}$$

Adding the discharge coefficient and substituting the velocity of approach factor into (8) and (9) the actual hydraulic equations are:

$$W = \left(\frac{\pi D_2^2}{4}\right)\left(\frac{C}{\sqrt{1-\beta^4}}\right)\left(2g\rho_1 \Delta P\right)^{1/2} \tag{11}$$

$$q = \left(\frac{\pi D_2^2}{4}\right)\left(\frac{C}{\sqrt{1-\beta^4}}\right) \tag{12}$$

Equation (11) now becomes

$$W = \left(\frac{\pi D_2^2}{4}\right)\left(K\right)\left(2g\rho_1 \Delta P\right)^{1/2} \tag{13}$$

The flow of compressible fluids through orifices and the relation between the compressibility and the discharge coefficients will now be considered.

With incompressible fluids and certain limitations upon orifice meter shapes

$$K = f_1 \left(N_{RE} , \beta , D \right) \tag{14}$$

For this discussion a particular pipe and orifice will be considered so that β and D are constants.

To define completely the nature of the flow of compressible fluids, the compressibility of the fluid Γ (Gamma) is needed in addition to the quantities D_2 , V_2 , μ and ρ . This factor can be defined by the relation

$$\Gamma = - \frac{1}{v} \frac{dv}{dP} \tag{15}$$

Hence, for a compressible fluid

$$K = f_2 \left(D_2 , V_{m_2} , \rho , \mu , \Gamma \right) \tag{16}$$

Since each of the five independent variables can be expressed in terms of three independent fundamental units, equation(16) can be reduced to an equivalent dimensionless relation having only (5-3) two independent variables or products, thus

$$K = f_3 \left(\pi_1 , \pi_2 \right) \tag{17}$$

let $$\pi_1 = D_2^x V_{m_2}^y \left(\frac{\rho}{g} \right)^z \mu \tag{18}$$

Then since the dimension of π_1 , is to be unity

$$\left[(L^{x})(L^{y}T^{-y})(F^{z}L^{-3z})(L^{-z}T^{2z})(FL^{-2}T)\right] = 1$$

Solving for X, Y, Z

$F;\quad Z + 1 = 0 \qquad \therefore Z = -1$

$T;\quad -Y + 2Z + 1 = 0 \qquad \therefore Y = 1$

$L;\quad X + Y - 3Z - Z - 2 = 0 \qquad \therefore X = -1$

Therefore:

$$\pi_1 = \frac{g\mu}{D_2 V_2 \rho} \tag{19}$$

Similarly let

$$\pi_2 = D_2^{X} V_{M_2}^{Y} \left(\frac{\rho}{g}\right)^{Z} \Gamma \tag{20}$$

Then $\left[(L^{x})(L^{y}T^{-y})(F^{z}L^{3z})(L^{-z}T^{2z})(F^{-1}L^{2})\right] = 1$

Solving for X, Y, Z

$F;\quad Z - 1 = 0 \qquad \therefore Z = 1$

$T;\quad -Y + 2Z = 0 \qquad \therefore Y = 2$

$L;\quad X + Y - 3Z - Z + 2 = 0 \qquad \therefore X = 0$

Therefore: $\pi_2 = V_{M_2}^{2} \frac{\rho}{g} \Gamma$

(21)

Substituting (20) and (21) into (17)

$$K = f_4\left(\frac{g\mu}{D_2 V_{m2}}, V_2^2 \frac{\rho}{g} \Gamma\right) \tag{22}$$

and

$$W = \left(\frac{\pi D_2^2}{4}\right)(NK)\sqrt{2g\rho\Delta P} \tag{13}$$

where N is a numerical constant.

Also

$$V_{M2} = \frac{4W}{\pi D_2^2 \rho}$$

substituting

$$V_{M2} = NK.\sqrt{\frac{2g\Delta P}{\rho}}$$

thus

$$V_{M_2}^2 \frac{\rho}{g} \Gamma = 2 N^2 K^2 \Delta P \Gamma \tag{23}$$

Rewriting equation (22)

$$K_1 = f_4\left(\frac{g\mu}{D_2 V_{m2} \rho}, 2N^2 K^2 \Delta P \Gamma\right) \tag{24}$$

which is equivalent to

$$K_1 = f_5\left(\frac{g\mu}{D_2 V_{m2} \rho}, \Delta P \Gamma\right) \tag{25}$$

Now if the assumptions are made that the fluid is an ideal gas and also that the flow is isentropic, so that

$$PV^K = \text{a constant} \tag{26}$$

Differentiating

$$V^K + PKV^{K-1}\left(\frac{\partial V}{\partial P}\right)_S = 0 \tag{27}$$

now

$$\Gamma = -\frac{1}{V}\left(\frac{\partial V}{\partial P}\right)_S \tag{28}$$

combining equations (27) and (28)

$$\Gamma = \frac{1}{\rho K} \tag{29}$$

Rewriting (25)

$$K_1 = f_5\left(\frac{g\mu}{D_2 V_{m2}\rho}, \frac{\Delta P}{\rho^K}\right) \tag{30}$$

by the general notation

$$X = \frac{P_1 - P_2}{P_1} = \frac{\Delta P}{P_1}$$

and

$$N_{RE} = \frac{D_2 V_{m2}\rho}{g\mu} \tag{31}$$

Rewriting equation (30)

$$K_1 = f_6\left(\frac{1}{N_{RE}}, \frac{X}{K}\right)$$

Knowing what K_1 is a function of, the values can be determined by experiment. If we were to plot K_1 vs X or X/K for various values of K and β, the results would be a family of lines with various slopes. Their equation would be:

$$K_0 = K_0 - \varepsilon\frac{X}{K} \tag{32}$$

$$K_1 = K_0\left(1 - \frac{\varepsilon}{K_0}\frac{X}{K}\right)$$

$$\text{Let}: \quad Y = \left(1 - \frac{\varepsilon}{K_0}\frac{X}{K}\right)$$

$$K_1 = K_0 Y_1 \tag{33}$$

Y_1 is termed the "net expansion factor" since it is introduced to take account of the effects of expansion as an expansible fluid flows through an orifice and the "hydraulic" equation is used for computing the rate of flow. K_0 is the limiting value of K_1 as $X = 0$. If $K_0 = K$, a relation which is often used without stating it, the hydraulic equation for use with a square edged orifice for compressible fluids becomes:

$$W = \left(\frac{\pi D_2^2}{4}\right) K Y_1 \sqrt{2g Y_1 \Delta P} \qquad (34)$$

Since ΔP is measured in inches of water

$$\Delta P = \Delta H_f \frac{(62.316)}{(1728)(144)}$$

also changing D_2 from feet to inches

$$W\,(lb/sec) = 0.0997\, D_2^2\, K Y_1 \sqrt{\rho_1 \Delta H_f} \qquad (35)$$

$$\rho = \frac{1}{14.1647}$$

$$W = 0.0997\, D_2^2\, (0.266)\, K Y_1 \sqrt{\Delta H_f}$$

$$= 0.02655\, D_2^2\, K Y_1 \sqrt{\Delta H_f} \qquad (36)$$

Equation (36) is the general form of the equation for flow of a compressible fluid through an orifice meter. More specific equations are necessary for use with the orifice opening than were used in the meter. The specific equations are as follows:

<u>For a 4 inch orifice</u>

$$W\,(lb/sec) = 0.425\, K Y_1 \sqrt{\Delta H_f} \qquad (37)$$

$$W\,(lb/hr) = 1530\, K Y_1 \sqrt{\Delta H_f} \qquad (38)$$

$$G\,(lb/hr.\text{-}ft^2) = W/3600\, A_2 = W/314.5 \qquad (39)$$

$$G = 4.86\, K Y_1 \sqrt{\Delta H_f} \qquad (40)$$

For a 3 inch orifice

$$W\,(\text{lb/sec}) = 0.2385\,K\,Y.\sqrt{\Delta H_1} \tag{41}$$

$$W\,(\text{lb/hr}) = 860\,K\,Y.\sqrt{\Delta H_1} \tag{42}$$

$$G\,(\text{lb/hr-ft}^2) = W/176.9 = 4.86\,K\,Y.\sqrt{\Delta H_1} \tag{43}$$

For a 2" orifice

$$W\,(\text{lb/sec}) = 0.1061\,K\,Y.\sqrt{\Delta H_1} \tag{44}$$

$$W\,(\text{lb/sec}) = 382\,K\,Y.\sqrt{\Delta H_1} \tag{45}$$

$$G\,(\text{lb/hr-ft}^2) = W/78.6 = 4.86\,K\,Y.\sqrt{\Delta H_1} \tag{46}$$

A set of curves relating manometer reading and W_A were calculated for each of the three orifice plates supplied. In the calculations, temperature and humidity of 100°F. and 50 grains respectively were assumed. The curves are presented as Figures 9, 10 and 11.

DEVELOPMENT OF CURVE FOR C_p

The values of specific heat were computed for a pound of wet air at varying values of temperature and specific humidity by the following equation where x is the specific humidity in grains per pound of dry air:

$$C_p = \frac{X(C_{p\,H_2O}) + 7000(C_{p_a})}{X + 7000} = \text{BTU/lb Wet Air}$$

The values of specific heat for wet air are plotted in Fig. 13.

In all calculations, the values used for viscosity and thermal conductivity are those given in Gas Tables by Keenan and Kaye, Ref. (12) for dry air. They are plotted for varying temperature in Fig. 12. Pierson, in Reference (15), states "The physical properties of air involved in the dimensionless groups have all been evaluated from data for dry air, investigation showing the effects of humidity to be negligible ...."

APPENDIX E

Sample Calculation

A sample calculation of data for a typical experimental point will be be given to illustrate the method employed.

The following experimental data was obtained:

Wet bulb temperature:		59.75^{o} F.
dry bulb temperature:		65.75^{o} F.
barometric pressure	:	29.75 "Hg
flow meter	:	4" orifice
$\triangle H_f$	:	20.25" H_2O
H_1	:	1.58 "Hg
T_1	:	92^{o}F.
T_2	:	147^{o}F.
tube temperature	:	215^{o}F.

1) $P_1 = (.4894)(1.58) = .77$ psi

2) barometric pressure $= (.4894)(29.75) = 14.55$ psia

3) $P = 14.55 + .77 = 15.32$ psia

4) $\Delta H_f / P = \dfrac{20.25}{15.32} = 1.322$

5) $Y_1 = .984$ from fig. 16

6) specific humidity $= 68 \dfrac{\text{grain}}{\text{lb. dry air}}$

7) $\mu = .0456$ at $T = 92^{o}$ F from Fig. 12

8) $\dfrac{D_2}{\mu} = \dfrac{(6.065)(3600)}{(12)(.0456)} = 39{,}800$

9) $4.86\sqrt{\Delta H_f}$ = 4.86 20.25 = 21.9

10) estimate K = .680

11) estimate R_e = (.984)(39,800)(21.9)(.680) = 583,000

12) K = .676 from fig. 17

13) $G = Y_1 \times 4.86\sqrt{\Delta H_f} \times K$

$= (.984)(21.9)(.676) = 14.60$

14) $W_A = 314.5\,G = (314.5)(14.60) = 4600$

Note: When fig. 11 is entered with the argument $\Delta H_f = 20.25$, W_A is determined to be 4610 lbs. per hour. Since the increase of accuracy obtainable by use of the above rigorous method is somewhat debatable, figure 11 will be used to obtain W_A. Since figure 11 was derived for standard conditions where $T_1 = 100^\circ$ F. and specific humidity = 50, any error will be small.

15) $\Delta T = T_2 - T_1 = 147 - 92 = 55^\circ$ F.

16) $T_{AVE} = \dfrac{T_1 + T_2}{2} = \dfrac{147 + 92}{2} = 119.5^\circ$ F.

17) for T = 119.5 and specific humidity = 68, C_p = .2422 from fig. 13

18) $Q = W_A C_p \Delta T = (4600)(.2422)(55) = 61300$ BTU/hr.

19) $D = LMTD = \dfrac{\Delta T}{\dfrac{T_t - T_1}{T_t - T_2}} = \dfrac{55}{\dfrac{215 - 92}{215 - 147}} = 92.5$

20) Effective heating surface:

$S = \dfrac{(220)(8.375)}{(4)(144)} = 10.04\ ft^2$

21) $U = \frac{Q}{SD} = \frac{61300}{(10.04)(92.5)} = 66.0$ BTU/ft^2 - hr. - $^\circ$F.

22) $T_f = \frac{T_t + T_{av}}{2} = \frac{215 + 119.5}{2} = 167^\circ$ F.

23) $k = .0174$ and $\mu = .0502$ from fig. 12

24) $N_{NU} = \frac{U\ D_t}{k} = \frac{66.0}{(48)(.0174)} = 79.1$

25) Free flow area:

$$A = \frac{(8.375)(8.375 - \frac{22}{4})}{144} = 0.1673 \text{ ft}^2$$

26) $N_{RE} = \frac{W_A\ D_t}{A\ \mu} = \frac{4600}{(48)(.1673)(.0502)} = 11440$

27) As a check, the value of NNU will be determined from the modified Grimison equation; for convenience, values of $.292\ F_A(R_e)^{.6}$ have been plotted in Fig. 14. $0.292\ F_A\ (N_{RE})^{0.6} = 84.2$

28) $\left(\frac{C_p \mu}{k}\right)^{1/3} = \left(\frac{.2422 \times .0502}{.0174}\right)^{1/3} = .886$

29) Since $F_d = 1.00$ for a tube bank ten rows deep, the theoretical Nusselt number is:

$$N_{NU} = .292\ F_A\ F_D\ (R_e)^{.6}\left(\frac{C_p \mu}{k}\right)^{1/3}$$

$$= (84.2)(.886)(1.00) = 74.6$$

Therefore the theoretical and experimentally determined values of Nusselt number agree within about six percent.

A sample friction factor calculation will be made for Point #1, Sheet #1:

1) $\Delta H_f = 20.62"\ H_2O$

$W_A = 4650$ lbs/hr from Fig. 11

2) $\Delta p = 8.88"\ H_2O$

3) $G = \frac{W}{A} = \frac{4650}{.1673} = 27800$ lbs/ft^2 - hr.

4) $T_t = 212^\circ$ F.

5) D = LMTD = 89.0 from corresponding calculation of N_{NU}

$.8D = .8(89.0) = 71.2$

6) $T_f = T_t - .8\ D = 212 - 71.2 = 140.8^\circ$ F.

7) $\mu = .0486$ from fig. 12

8) $N_{RE} = \frac{GDt}{\mu} = \frac{(27800)}{(48)(.0486)} = 11900$

9) $G^2 = (27800)^2 = 7.71 \times 10^8$

10) $\rho = \frac{P}{RT} = \frac{(15.0)(144)}{(53.3)(460 + 140.8)} = .0672$ lbs/ft^3

11) $f = \frac{10.84 \times 10^8}{NG^2} = \frac{(10.84)(.0672)(8.88)}{(10)(7.71)} = .0841$

APPENDIX F - CHARTS

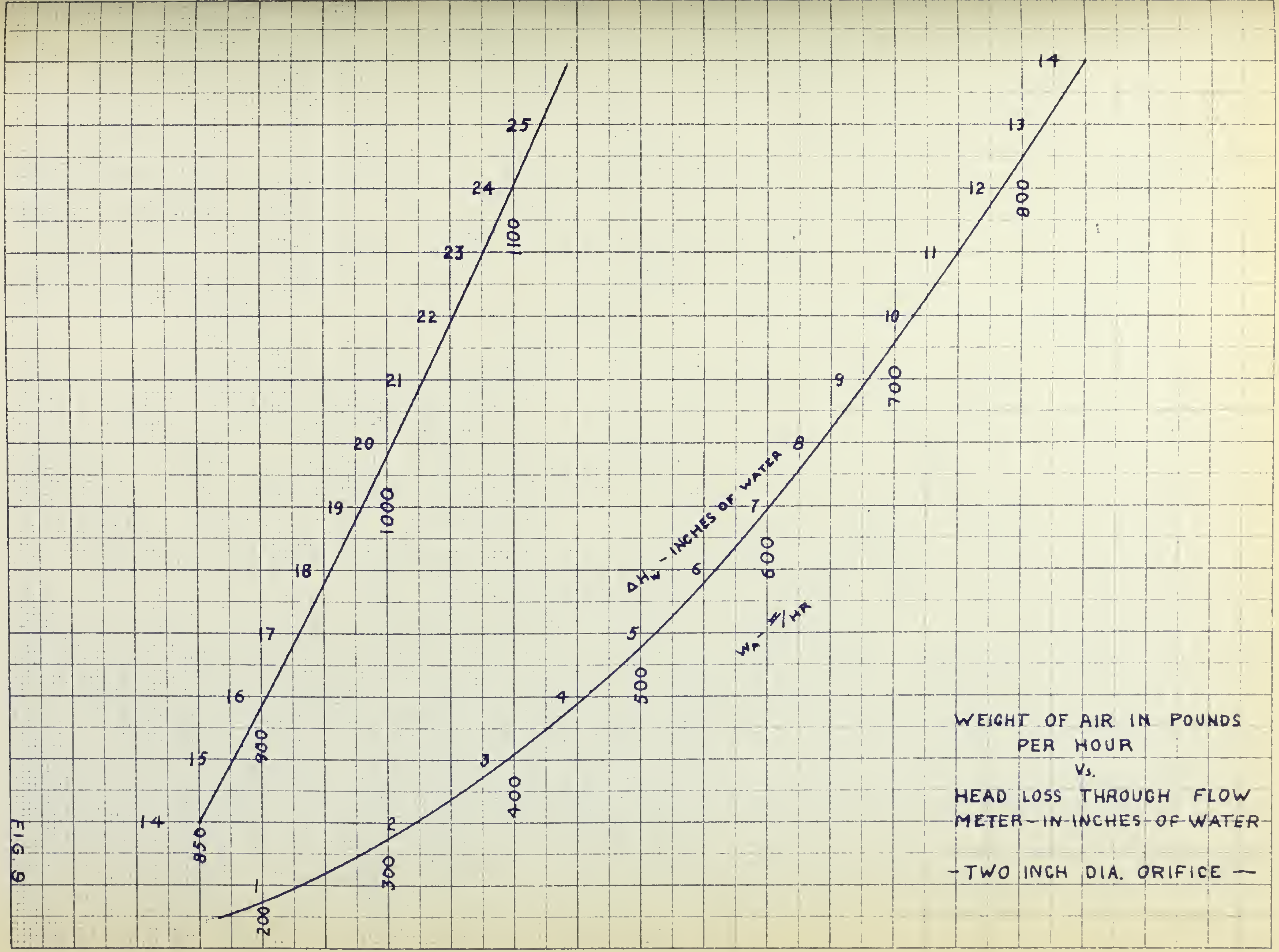

WEIGHT OF AIR IN POUNDS PER HOUR
Vs.
HEAD LOSS THROUGH FLOW METER — IN INCHES OF WATER
— TWO INCH DIA. ORIFICE —
ΔH_w - INCHES OF WATER
W_A - #/HR
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23
24
25
200
300
400
500
600
700
800
850
900
1000
1100

FIG. 9

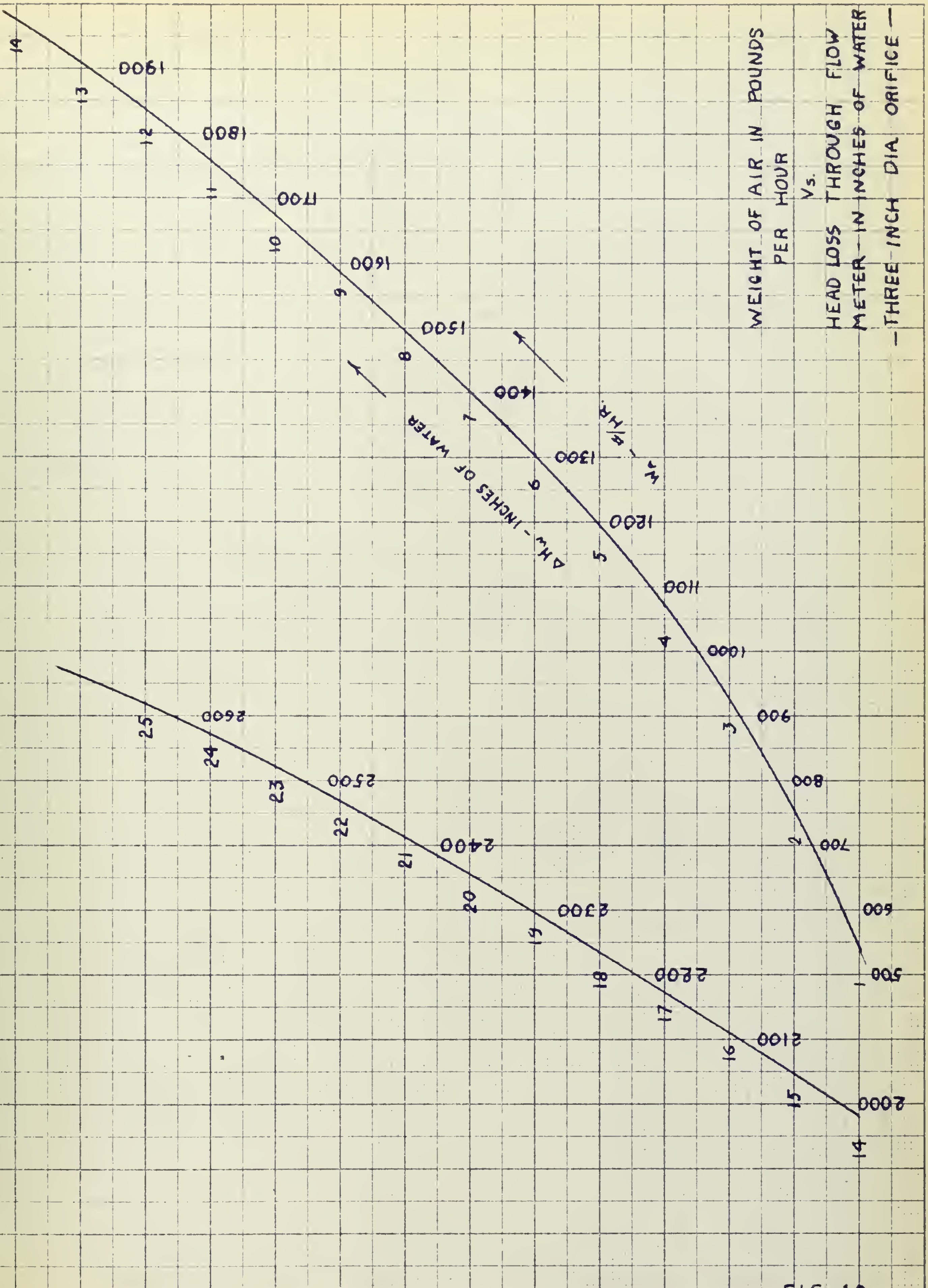

WEIGHT OF AIR IN POUNDS PER HOUR
Vs.
HEAD LOSS THROUGH FLOW METER - IN INCHES OF WATER
- THREE INCH DIA ORIFICE -
ΔHw - INCHES OF WATER
W - #/HR.

FIG. 10

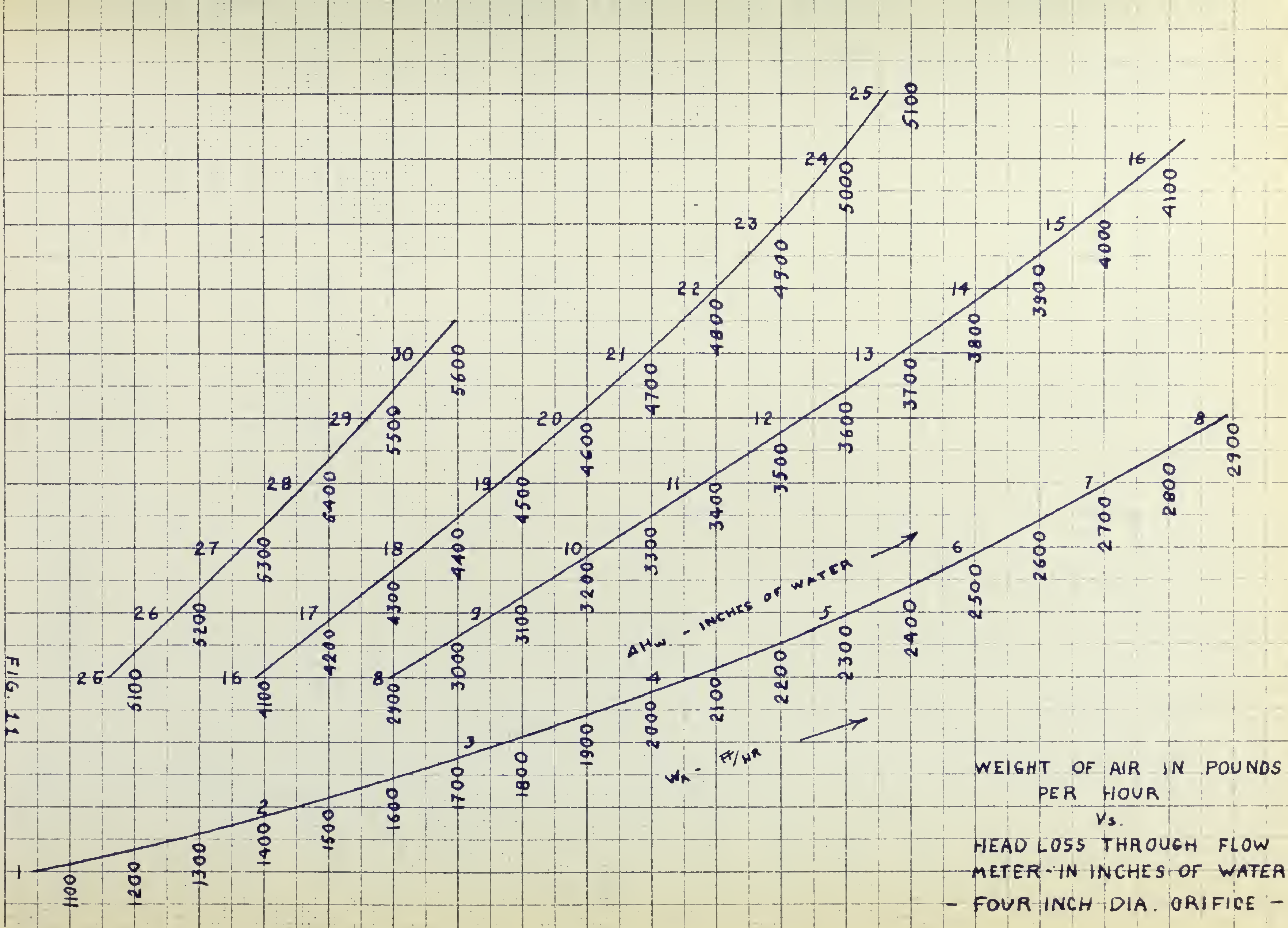

WEIGHT OF AIR IN POUNDS PER HOUR
Vs.
HEAD LOSS THROUGH FLOW METER IN INCHES OF WATER
- FOUR INCH DIA. ORIFICE -
ΔHw - INCHES OF WATER
Wa - #/HR

FIG. 11

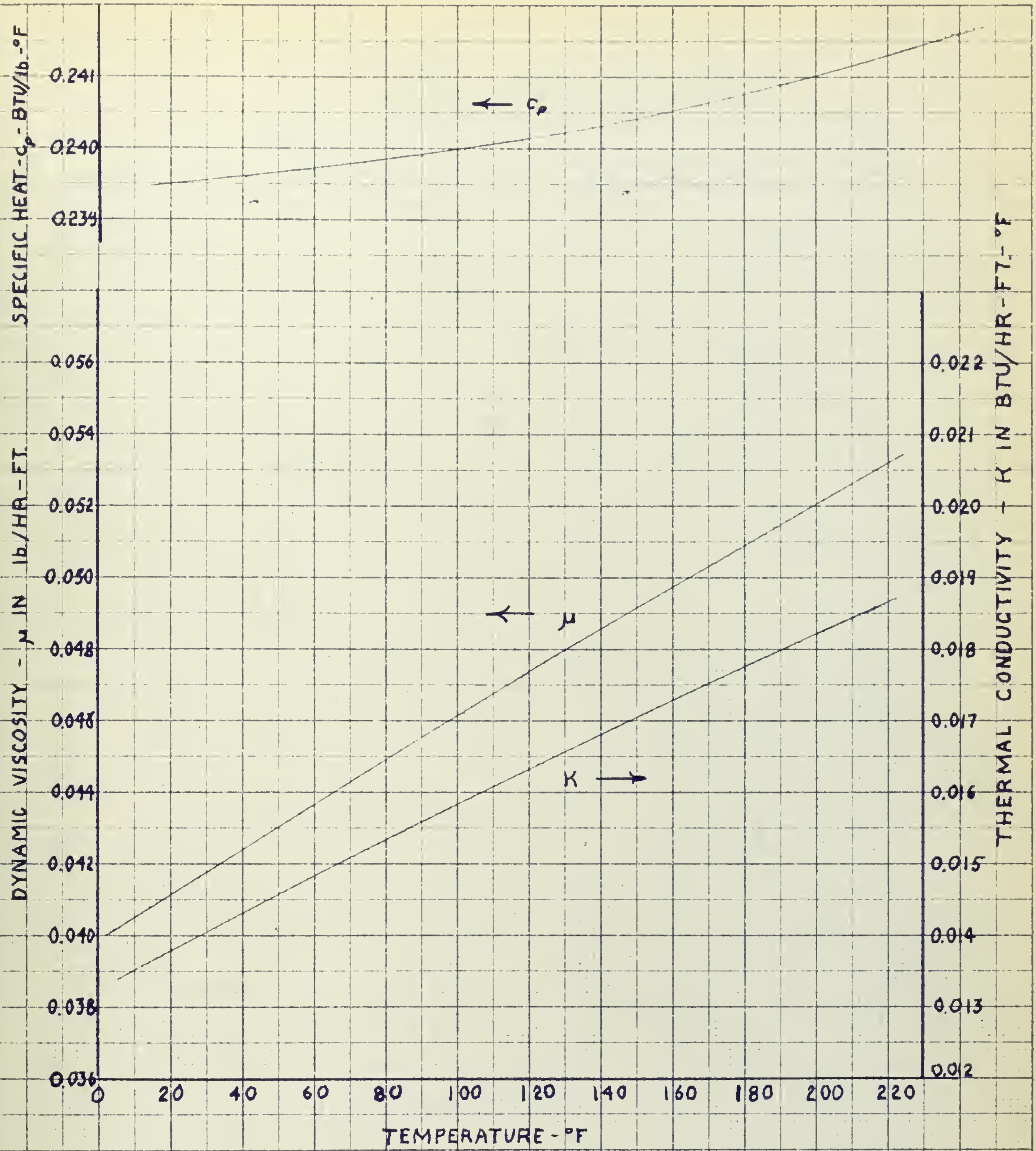


FIGURE 12

PROPERTIES OF DRY AIR

FROM: KEENAN AND KAYE, GAS TABLES

SPECIFIC HEAT OF AIR WITH MOISTURE
VALUES ARE PER lb. WET AIR
COMPUTED FROM VALUES OF SPECIFIC
HEAT FOR DRY AIR AND WATER VAPOR
FROM; "GAS TABLES", KEENAN & KAYE

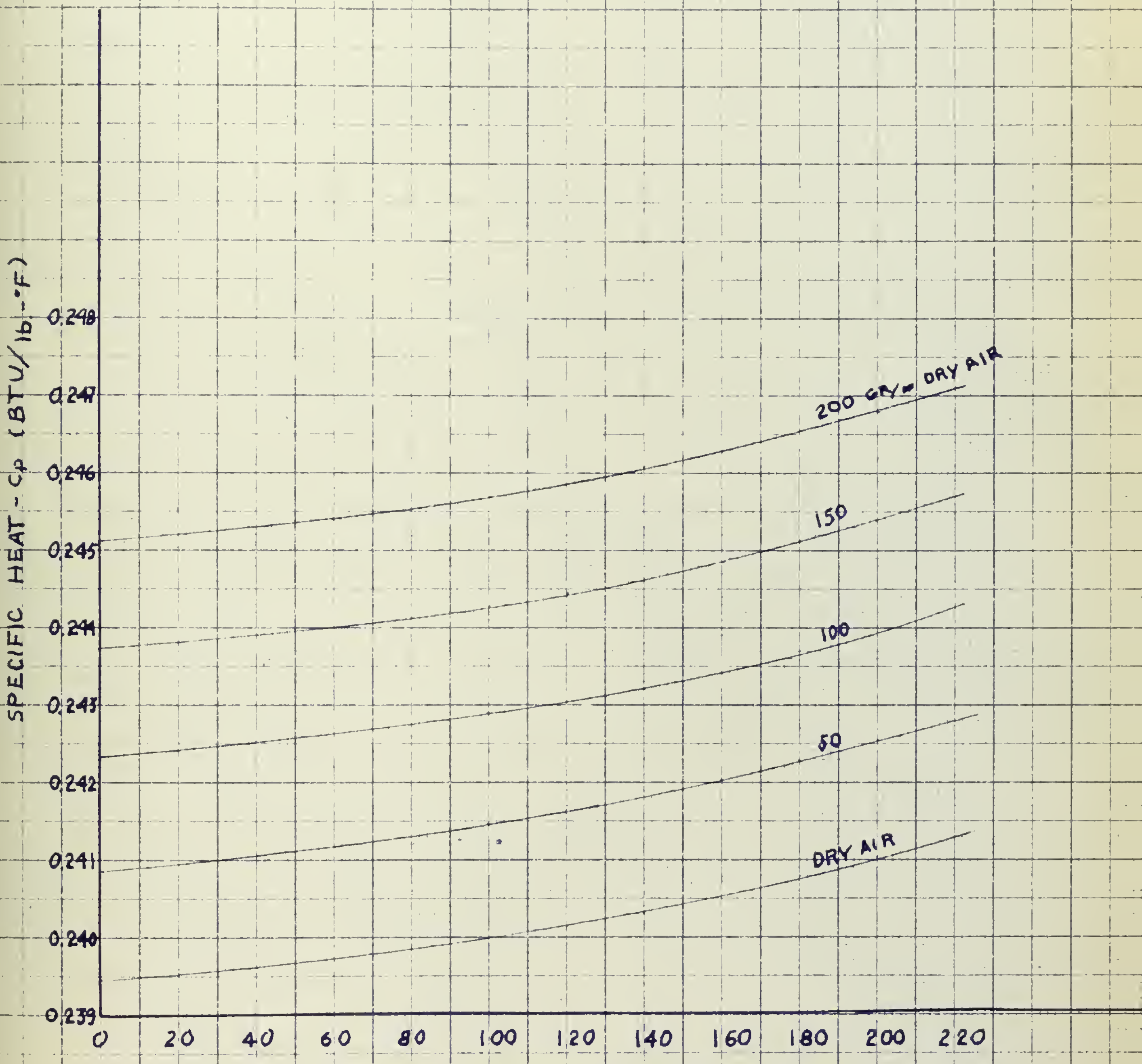
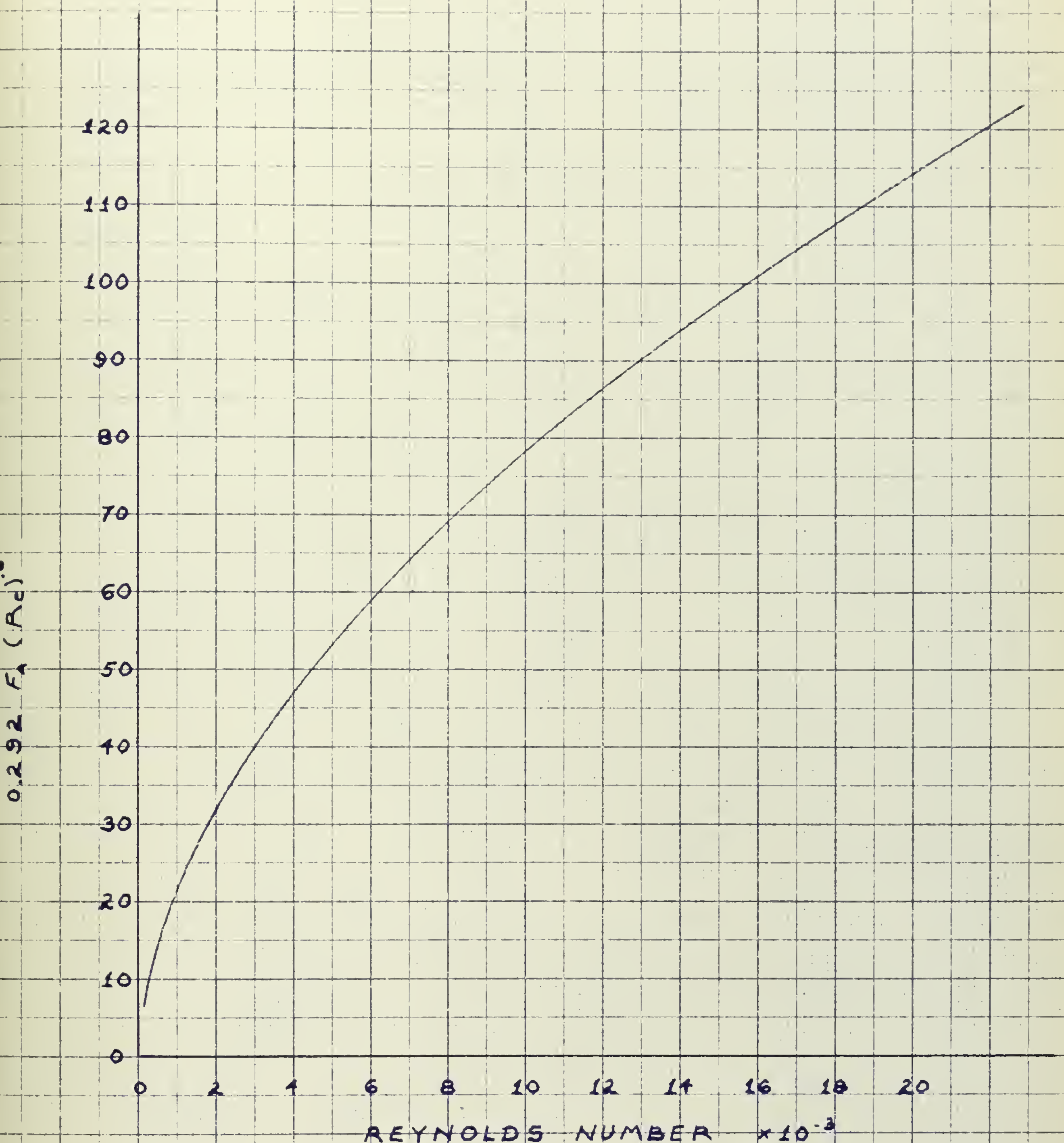
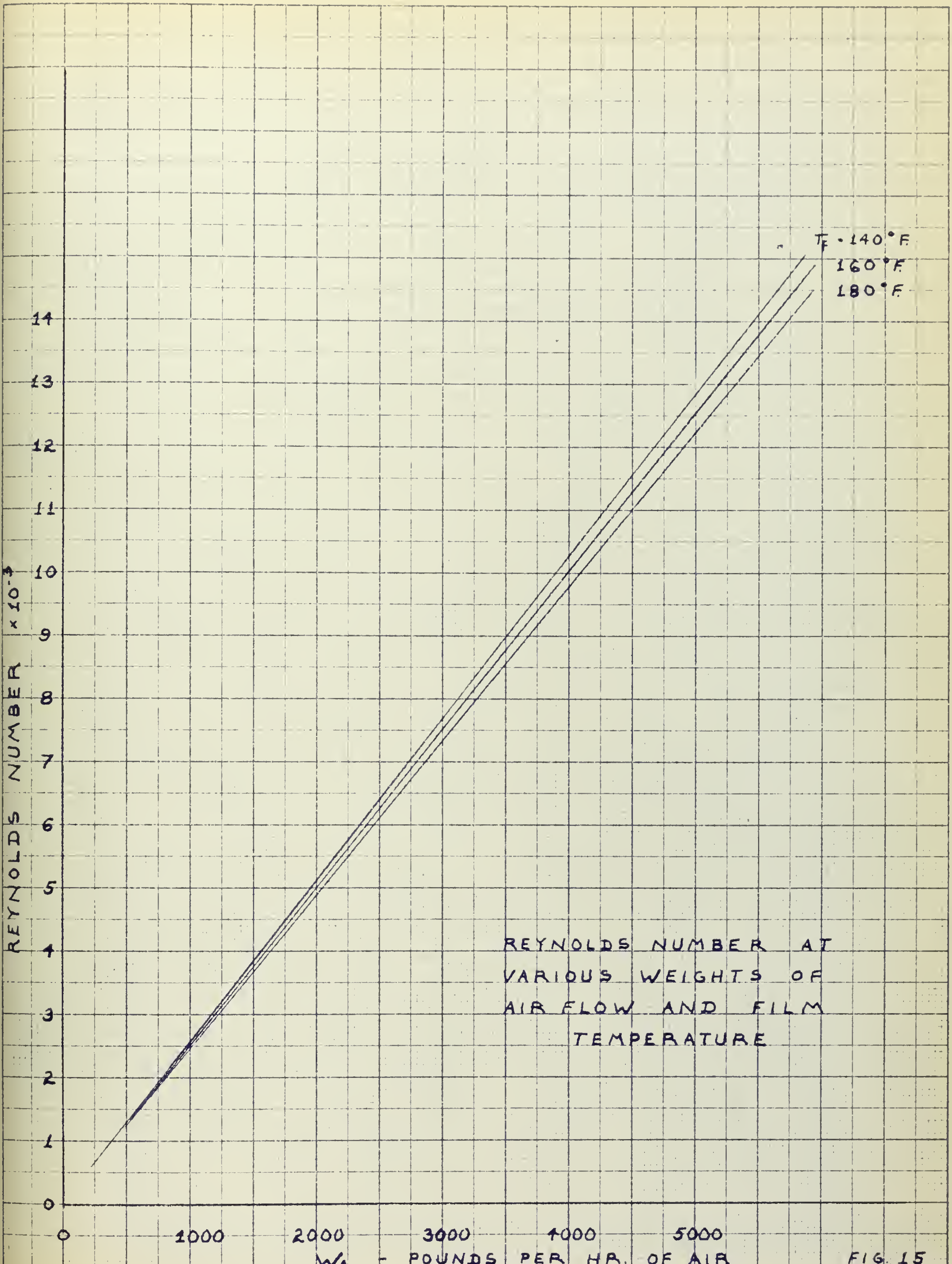


FIGURE 13



$0.292\ F_2\ (R_e)^{.6}$ VS. REYNOLDS NUMBER
WHERE $\frac{S_T}{D} = \frac{S_L}{D} = 1.5$

FIG. 14



T_F = 140°F
160°F
180°F
REYNOLDS NUMBER × 10^{-3}
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
0
1000
2000
3000
4000
5000
W_A - POUNDS PER HR. OF AIR
REYNOLDS NUMBER AT VARIOUS WEIGHTS OF AIR FLOW AND FILM TEMPERATURE

FIG. 15

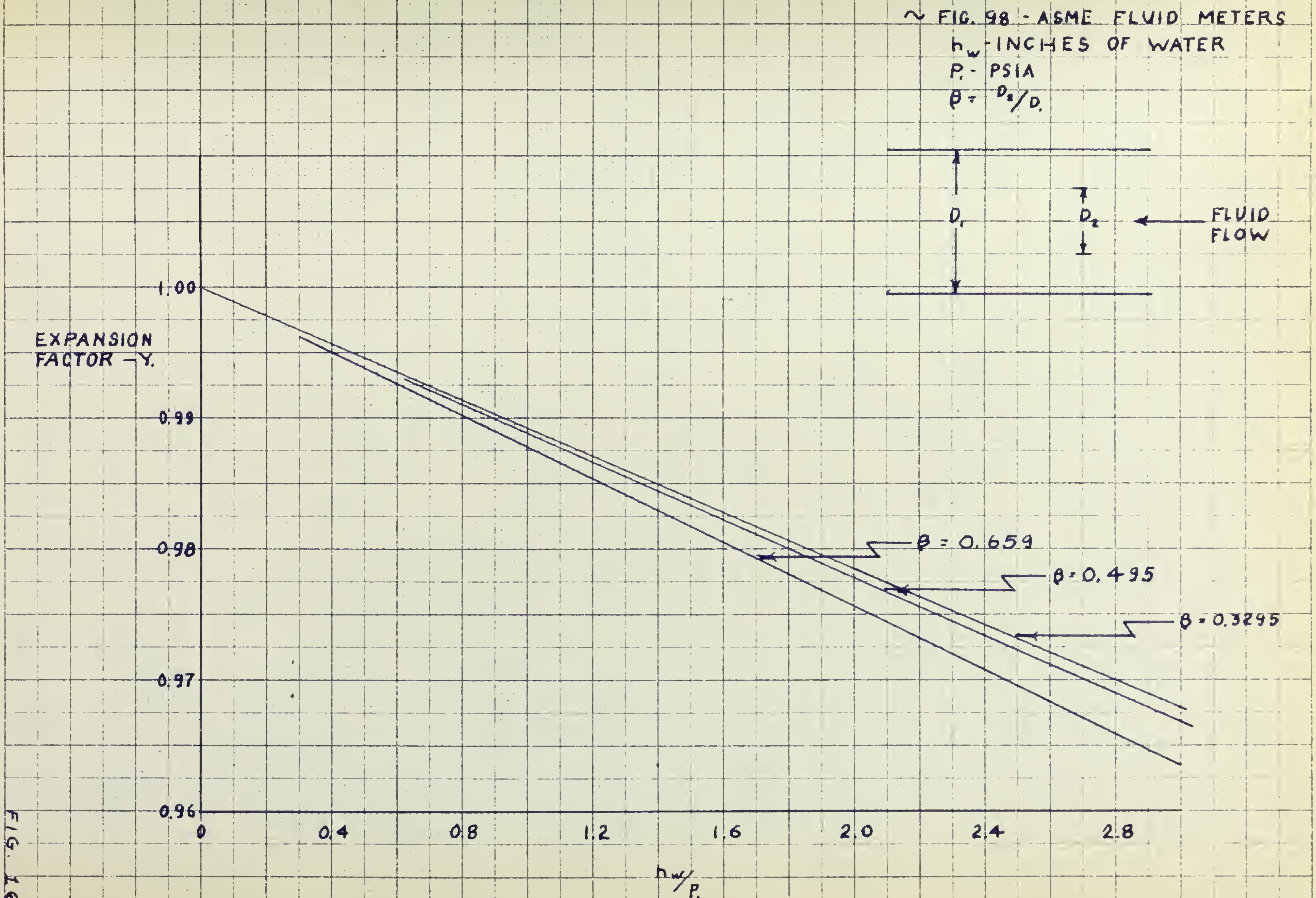

~ FIG. 98 - ASME FLUID METERS
h_w - INCHES OF WATER
P_1 - PSIA
$\beta = D_2/D_1$
D_1
D_2
FLUID FLOW
EXPANSION FACTOR — Y.
1.00
0.99
0.98
0.97
0.96
0
0.4
0.8
1.2
1.6
2.0
2.4
2.8
h_w/P_1
$\beta = 0.659$
$\beta = 0.495$
$\beta = 0.3295$

FIG. 16

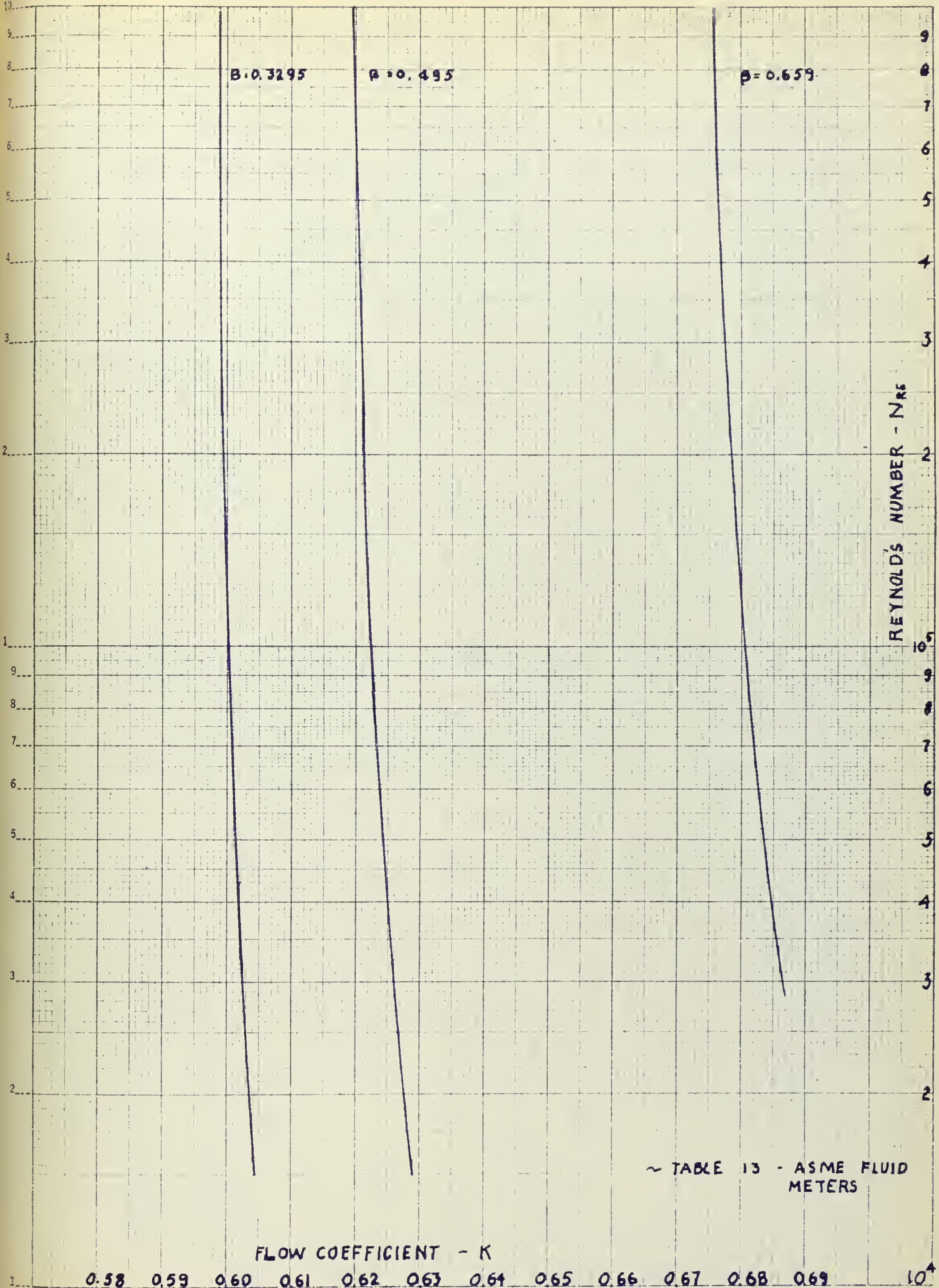

B=0.3295
β=0.495
β=0.659
REYNOLDS NUMBER - N_{RE}
10^5
10^4
~ TABLE 13 - ASME FLUID METERS
FLOW COEFFICIENT - K
0.58
0.59
0.60
0.61
0.62
0.63
0.64
0.65
0.66
0.67
0.68
0.69

FIG. 17

APPENDIX F

CALCULATION SHEETS FOR CALCULATION OF NUSSELT NUMBER AND FRICTION FACTOR

APPENDIX F

Calculation Sheet No. 1

Specific Humidity: 50 $\frac{\text{grains}}{\text{lb. dry air}}$

Point	2	3	4
ΔH_f	20.5	18.0	9.0
W_a	4640	4340	3060
T_2	145	145	144
T_1	91	89	76
ΔT	54	56	68
T_{av}	118	117	120
T_t	208	210	210
diff	90	93	90
T_f	163	163	165
C_p (@T_{av})	.2416	.2416	.2416
K (@T_f)	.0174	.0174	.0174
μ_a (@T_f)	.050	.050	.050
$\ln \frac{T_t - T_1}{T_t - T_2}$	.618	.620	.708
D = LMTD	87.2	90.3	96.0
$Q = WC_p \Delta T$	60500	58600	50200
$U = \frac{Q}{(10.04)\,D}$	69.1	64.7	52.1
$R_e = \frac{W}{(8.04)\mu}$	11540	10820	7620
$N_{Nu} = \frac{U}{48k}$	82.7	77.5	62.4
$\left(\frac{C_p \mu}{k}\right)^{1/3}$	.886	.886	.886
$.292(R_e)^{.6} F_a F_d$	84.5	82.0	67.3
N_{Nu} (theor.)	74.9	72.7	59.7

APPENDIX F

Calculation Sheet No. 2

Specific Humidity: 68 $\frac{\text{grains}}{\text{lb. dry air}}$

Point	1	2	4	5	6	7
ΔH_f	20.25	17.25	7.88	6.25	5.50	3.88
W_a	4610	4250	2870	2570	2420	2030
T_2	147	146	149	150	151	152
T_1	92	90	80	78	77	76
ΔT	55	56	69	72	74	76
T_{av}	119.5	118	114.5	114	114	114
T_t	215	210	210	212	215	215
diff	95.5	92	95.5	98	101	101
T_f	167	164	162	163	164	164
C_p (@T_{av})	.242	.242	.242	.242	.242	.242
K(@T_f)	.0174	.0174	.0174	.0174	.0174	.0174
μ_a (@T_f)	.0502	.0500	.0500	.0500	.0500	.0500
$\ln \frac{T_t - T_1}{T_t - T_2}$	.593	.629	.755	.770	.765	.792
D = LMTD	92.5	89.1	91.2	93.5	96.7	95.8
$Q = WC_p\Delta T$	61300	57600	47900	44800	43300	37400
$U = \frac{Q}{(10.04)\ D}$	66.0	64.4	52.3	47.7	44.6	38.9
$R_e = \frac{W}{(8.04)\mu}$	11440	10600	7160	6410	6040	5060
$N_{NU} = \frac{U}{48k}$	79.1	77.1	62.6	57.1	53.5	46.6
$\left(\frac{C_p \mu}{k}\right)^{1/3}$	.886	.886	.886	.886	.886	.886
$.292(R_e)^{.6}\ F_aF_d$	84.2	81.0	65.0	61.0	59.3	53.5
N_{Nu} (theor.)	74.6	71.8	57.6	54.1	52.6	47.4

APPENDIX F

Calculation Sheet No. 3

Specific Humidity: 72 $\frac{\text{grains}}{\text{lb. dry air}}$

Point	1	2	3	4	5	6	7
ΔH_f	20.62	17.12	12.62	11.12	5.50	4.75	1.62
W_a	4650	4230	3625	3400	2420	2250	1310
T_2	145	145.5	146.5	149	153	153	161
T_1	96.5	94	89.5	88	83	82.5	82
ΔT	48.5	51.5	57	61	70	70.5	79
T_{av}	120.8	119.8	118	118.5	115	117.8	121.5
T_t	212	212	212	212	212	212	212
diff	91.2	92.2	94	93.5	97	94.2	90.5
T_f	166.4	165.8	165	165.3	163.5	164.8	166.7
C_p (@T_{av})	.2423	.2423	.2423	.2423	.2422	.2423	.2424
K (@T_f)	.0174	.0174	.0174	.0174	.0174	.0174	.0174
μ_a (@T_f)	.0501	.0501	.0500	.0500	.0499	.0500	.0501
$\ln \frac{T_t - T_1}{T_t - T_2}$	.545	.574	.626	.678	.784	.786	.936
D = LMTD	89.0	89.8	91.0	90.0	89.3	89.7	84.4
$Q = WC_p\Delta T$	54600	52750	50100	50200	41000	38300	25100
$U = \frac{Q}{(10.04)\ D}$	61.1	58.5	54.8	55.5	45.7	42.5	29.6
$R_e = \frac{W}{(8.04)\mu}$	11550	10520	9030	8470	6040	5600	3250
$N_{Nu} = \frac{U}{48k}$	73.0	70.0	65.5	66.3	54.6	50.8	35.4
$\left(\frac{C_p\mu}{k}\right)^{1/3}$	.886	.886	.886	.886	.886	.886	.886
$.292(R_e)^{.6}\ F_aF_d$	84.7	80.2	74.0	71.5	59.5	56.8	41.5
N_{Nu} (theor.)	75.1	71.0	65.6	63.4	52.7	50.4	36.8

APPENDIX F

Calculation Sheet No. 4

Specific Humidity: 85 $\frac{\text{grains}}{\text{lb. dry air}}$

Point	1	2	3	4	5	6
ΔH_f	24.62	17.0	10.62	7.12	3.75	1.75
W_a	2600	2175	1728	1414	1040	704
T_2	155.5	157	161	164.5	169	170.5
T_1	90.5	88	85.5	84	83	82
ΔT	65	69	75.5	80.5	86	88.5
T_{av}	123	122.5	123.2	124.2	126	126.2
T_t	212	212	212	212	212	212
diff	89	89.5	88.8	87.8	86	85.8
T_f	167.5	167.2	167.6	168.1	169	169.2
$C_p(@T_{av})$	.2427	.2427	.2427	.2427	.2428	.2428
$K\ (@T_f)$	.0176	.0176	.0176	.0177	.0177	.0177
$\mu_a\ (@T_f)$	.0506	.0506	.0506	.0507	.0508	.0508
$\ln \frac{T_t - T_1}{T_t - T_2}$	.765	.813	.908	.990	1.098	1.140
D = LMTD	85.0	84.8	83.2	81.3	78.3	77.7
$Q = WC_p\Delta T$	41100	34900	31650	27650	21750	15160
$U = \frac{Q}{(10.04)\ D}$	48.1	41.0	37.9	33.9	27.6	17.6
$R_e = \frac{W}{(8.04)\ \mu}$	6380	5340	4240	3480	2550	1723
$N_{Nu} = \frac{U}{48k}$	55.8	47.6	43.6	39.9	32.5	20.7
$\left(\frac{C_p\ \mu}{k}\right)^{1/3}$	.888	.888	.888	.888	.888	.888
$.292(R_e)^{.6}F_aF_d$	60.8	55.0	48.4	43.5	36.8	29.0
N_{Nu} (theor.)	53.9	48.8	42.9	38.6	32.6	25.7

APPENDIX F

Friction Factor Calculation Sheet 1

Friction Factor $(f) = \dfrac{10.84 \times 10^{8} \rho \Delta P}{NG^{2}}$

Point	1	2	3	4	5	6	7
ΔH_f	20.62	17.12	12.62	11.12	5.50	4.75	1.62
W_a	4650	4230	3625	3400	2420	2250	1310
Δp ("H_2O)	8.88	7.38	6.25	5.12	2.64	2.38	0.88
$G = \frac{W}{A}$	27800	25300	21650	20300	14460	13450	7830
T_t	212	212	212	212	212	212	212
.8 D	71.2	71.8	72.8	72	71.5	71.8	67.5
T_f	140.8	140.2	139.2	140	140.5	140.2	144.5
μ	.0486	.0486	.0485	.0486	.0486	.0486	.0488
$R_e = \frac{G}{4\delta}$	11900	10840	9310	8700	6200	5770	3340
G^2 $(X10^{-8})$	7.71	6.40	4.69	4.13	2.09	1.81	.613
ρ	.0672	.0672	.0672	.0672	.0672	.0672	.0672
f	.0841	.0842	.0974	.0906	.0922	.0957	.1042

N = number of tube rows = 10

APPENDIX F

Friction Factor Calculation Sheet 2

Friction Factor (f) = $\dfrac{10.84 \times 10^{8} \rho \Delta P}{NG^2}$

Point	1	2	3	4	5	6
ΔH_f	24.62	17.0	10.62	7.12	3.75	1.75
W_a	2600	2175	1728	1414	1040	704
Δ p ("H_2O)	2.9	2.23	1.435	1.01	.55	.312
$G = \frac{W}{A}$	15530	13000	10320	8450	6220	4210
T_t	212	212	212	212	212	212
.8 D	68	67.8	66.6	65.1	62.6	62.2
T_f	144	144.2	145.4	146.9	149.4	149.8
μ	.0488	.0488	.0489	.0490	.0491	.0492
$R_e = \frac{G}{48}$	6640	5550	4410	3600	2635	1784
G^2 ($\times 10^{-8}$)	2.41	1.69	1.07	.714	.387	.177
ρ	.0672	.0672	.0672	.0672	.0672	.0672
f	.0878	.0963	.0980	.1032	.1038	.1286

N = number of tube rows = 10

APPENDIX G

Original Apparatus Using Water as a Cooling Medium

Table of Contents:

Development, Description, and Calibration of Original Apparatus Using Water as a Cooling Medium

Introduction

As often happens in the process of development, the final product of this investigation bore little resemblance to the original apparatus designed and constructed by Mulford and Graap (7). The original apparatus was experimented with, altered, and finally discarded. The experience gained was valuable in the design of the final apparatus, however, so a description of the method of approach, the problems involved, and the conclusions drawn are included in this appendix.

A complete description of the original apparatus is contained in Reference (7). The various modifications to the equipment and the reasons for making them are described here.

GENERAL DEVELOPMENT OF APPARATUS

The authors commenced work using the apparatus designed and constructed by Mulford and Graap, Webb '61. This apparatus was used, with some modifications, throughout. There were a number of undesirable features of the apparatus which it seemed desirable to attempt to correct at the outset. Mulford and Graap had been unable to seal the test section adequately so that there was an unacceptable amount of leakage of water along the tubes and outward toward the headers. They had felt that an arrangement of "water boxes" could be devised to contain the leakage and perhaps to stop it by application of a balancing air pressure. This arrangement was considered to be unwieldly, and it was decided to attempt to improve on the type metal seals. It was noted that the tube surfaces in way of the seals had not been tinned; it was felt that this lack of tinning reduced the likelihood of any sort of metallic seal adhering to the copper tubes sufficiently to preclude water leakage.

The apparatus was dismantled and the tube bank removed to the Machine Shop. The type-metal seals at either end of the tube bank were melted out with an acetylene torch. At this stage, the undesirability of the five-header arrangement (two rows of tubes to each set of headers) was first noted. In order to dismantle the bank two rows at a time, it was necessary to apply an excessive amount of heat which resulted in some warping of the tubes due to uneven heating. It was also extremely difficult to remove all traces of type metal from the tubes where the seals had been due to poor accessibility between the rows of tubes.

The tubes were separated and cleaned as much as was practicable. The

tubes were then pickled and the center section of each tube pair was tinned with solder at the New York Naval Shipyard.

Two attempts were made at pouring end seals with a mixture of 50-50 solder and type metal. The second attempt was successful. The tube bank was assembled, and dams of asbestos cement were included at either end packed between and around the individual tubes to retain the solder. The bank was then clamped together in the assembled position. A separate pour was made for each end seal with the bank in a sand mold with the tubes vertical. The tubes were heated with an acetylene torch preparatory to pouring and then sprayed with muriatic acid. The torch was played on the tubes during the pouring in both cases. The quantity of metal required made it necessary to use two ladles and resulted in a "layered" or segregated effect in the seals. After the second seal was poured, a noticeable warping of some tubes in the bank was discovered. It is believed that this condition resulted from the necessarily uneven heating of the tubes with the acetylene torch and subsequent cooling of tubes with both ends confined. Visual inspection and a hose test indicated satisfactory sealing on each end, so the tube bank was placed in the test section.

The test section was reassembled and fitted. Canvas gaskets were used to make the rear cover watertight and were soaked with permatex. Oakum was used to calk seams in the top and bottom. Angle iron strongbacks were installed to be drawn up by the assembly bolts. A U-tube mercury manometer was installed across the test section to measure the pressure drop.

At this juncture, it appeared that the originally designed tube bank left something to be desired as regards flexibility and ease of sealing. It was decided to design a new bank for concurrent fabrication while the original bank was being tested. Consequently a new bank was designed and drawn incorporating the improvements learned to date. The size of the steam headers was increased. The individual tube banks were single; rows of tubes rather than two row sections.

The test section was closed up and the circulating pump started. The apparatus was tested under full operating water pressure of about 24" of water. Relatively slight leakage was noted and was considered to indicate satisfactory tightness. One leak through the upper seal was found. A drip pan was installed under the test section with a line and nipple leading back to the circulating tank to return the leakage direct to the tank.

It was found that the capacity of the circulating pump was considerably less than the water flow for which the apparatus had been designed. The maximum obtainable flow was about 115 gpm, which corresponds to a reading of $4\frac{1}{2}$" Hg on the orifice meter. At this flow rate, the pump motor was overloaded, drawing 35 amps instead of the maximum permissible 30 amps. Inasmuch as the pump discharge pressure was measured at 60 psig, this low flow rate was deduced to indicate unacceptably large frictional losses in the line.

A 2" discharge line was substituted for the old $1\frac{1}{2}$" one and a smaller 11" O.D. impeller installed in the circulating pump. After these modifica-

tions, pumping capacity was found to be 221 gpm. The pump motor drew 30 amps and operated at 1700 rpm.

During the course of attempting to operate the apparatus, it was noticed that the circulating water tank was extremely rusty. Large particles of rust and debris were pumped into the system where they tended to accumulate in the tube bank. The tank was drained, cleaned, wire brushed, and painted with a total of four coats of Rustoleum primer. It is apparent that the hot water is detrimental to most usual means of preservation, since the four coats of primer failed after only a short period of service.

The apparatus was instrumented with mercury-in-glass thermometers and iron-constantan thermocouples. A pressure gage was installed on the steam separator. Mercury manometers were installed on the steam inlet headers to measure steam pressure drop across the tube bank. It was intended to measure water temperature entering and leaving the test section by both thermometer and thermocouple and to measure entering and leaving steam temperatures by means of thermocouples inserted into the respective headers. Thermocouples were read by a slide wire potentiometer across the thermocouple circuit. Water temperature measurements by thermocouple were found to be identical with direct readings on thermometers in the same positions. Moreover, it was found to be impracticable to obtain more than two or three thermocouple readings per minute because of the excessive amount of time involved in adjusting the potentiometer to obtain a reading. Iron-constantan thermocouples gave unsatisfactory performance in measuring steam temperature, because they corroded rapidly in the moist environment to which they were

exposed in the headers. An attempt was made to measure tube temperature above and below the test section by means of thermocouples placed in close proximity with tubes in the interior of the bank. The mechanical contact did not give a sufficiently accurate indication of temperature of the tubes, and the thermocouples eventually failed due to corrosion.

Some additional equipment was added to the apparatus for purposes of calibration. A dump line was installed in the test section discharge line direct to the laboratory drain trough, and a steam jacketed kettle was mounted and connected so that water could be heated to a boiling temperature and dumped quickly into the circulating tank. The system was then calibrated as described in a later section.

A series of test runs were made with poor results. The rate of heat transfer varied from approximately one fifth to one fourth of the theoretical rate. Three methods of taking and calculating data were employed; instantaneous temperature rise across the bank, temperature rise under steady state conditions using the cooling tower, and time rate of temperature rise. All three methods gave almost uniformly poor results, so that faulty design of the equipment was suspected. Five thermometers were placed in a vertical line immediately following the test section for comparison of their readings. The temperature indications of the bottom two were found to be appreciably lower than the temperatures of the top three.

Two possible causes of this phenomenon were postulated: either drainage of the tubes was inadequate, or steam supply was insufficient to heat them evenly. Either cause was a possibility, since the cross section area of

each header was less than the sum of the cross section areas of the tubes butted into it. In the former case it was suspected that the steam was condensing in the tubes leaving them half filled with water. The ends were cut off the exhaust headers and they were butted into two copper exhaust boxes connected to large $1\frac{1}{2}$" discharge lines draining into the drain trough. It is believed that this remedy precluded the possibility of the tubes being water bound. A copious quantity of condensate and steam was observed emanating from the tube bank. Additional steam supply lines of 3/8" copper tubing were run to each of the five steam supply headers. Further tests showed all five thermometers immediately following the tube bank to be reading fairly consistently. Consequently, it was believed that the problem of temperature stratification was solved.

Calculations based on data obtained indicated that the heat transfer rate was only about one half of the theoretical value based on the modified Grimison equation. It was thought that a film had built up on the tube watersides sufficient to reduce the heat transfer and cause the apparent discrepancy. It was then decided to clean the tube bank thoroughly and then to use clean, cold water for each subsequent test run. The circulating tank was filled with clean water; and detergent, caustic soda, and ammonia were added. The mixture was circulated through the test section for about two hours. It is recommended that future investigators perform this experiment at least once -- The suds rise to a magnificent height, and the entertainment value is considerable. The test section was then rinsed

and drained and cleaned further with a steam lance.

After cleaning, the tube bank was instrumented with chromel-alumel thermocouples in two locations. The test section was then reassembled and data taken again under test conditions. Measured tube temperatures were used in calculating the heat transfer rate. The results were again unsatisfactory in that correlation with theoretical data was not obtained. Tube temperatures obtained by thermocouple measurement were approximately 130 degrees F., or less than the exit water temperature at the end of the test run. It was believed that the thermocouples were installed improperly to measure the true tube wall temperature and were being influenced unduly by the temperature of the water flowing over the soldered hot junctions. Two additional methods of thermocouple installation were tried. First the hot junctions were soldered into grooves cut into the tube walls, then they were soldered into small holes drilled into the tube walls. Both of these methods of installation produced results identical to the first case in which the hot junctions were merely soldered to the tube surface. It was concluded that some experimentation with thermocouple installation was necessary to ensure better results.

Thermocouples were attached to a 3/8" copper tube by each of the foregoing methods. In addition, one thermocouple was attached so that its hot junction was covered by a piece of $\frac{1}{4}$" tubing. The tube with thermocouples attached was connected to a steam supply line and partially immersed in cold water. With the steam turned on and the thermocouple junctions under

water, the tube temperatures indicated by each of the thermocouples were compared. All indicated tube temperatures were 212 degrees F. This result was expected since the saturated steam supply was so regulated as to produce partial condensation inside the tube. The temperatures were then noted when the cold water surrounding the tube was agitated. The temperatures indicated by all thermocouples dropped radically. By vigorous agitation it was possible to reduce the tube temperature to as little as 135 degrees F. This experiment seemed to indicate that heat transfer from the copper to the water increased in direct proportion to relative velocity of the water (Reynolds Number), but that heat transfer from the condensing steam to the copper was insufficient to maintain the tube temperature constant. This finding was borne out by a comparison of the relative conductances of water and steam films.

Therefore, the validity of a primary assumption upon which design of the test apparatus had been based was disproved. Specifically, the premise that the temperature of tubes containing condensing steam is constant at 212 degrees F. was shown to be incorrect under the conditions encountered in the operation of this apparatus. Further, no accurate method of measuring tube temperature was found during the investigation. It was felt that the low values of tube temperature obtained by the crude thermocouple configurations employed were useless for calculations; certainly they did not produce results which could be correlated with data obtained previously by other investigators.

Several considerations concerning future action suggested themselves.

A satisfactory means of measuring tube surface temperature had not been found after some amount of work, and no immediate solution to the problem was at hand. Further, previous investigators have had much difficulty with the same problem. Previous investigators working on experimental determination of heat transfer at Webb Institute have had considerable success in working with air as the heated fluid. The apparatus was readily adaptable to the use of air.

It was decided to test the performance of the apparatus with air. The circulating water system was disconnected, and the turbo blower in the laboratory was connected to the orifice meter. The apparatus was tested and the experimental heat transfer rate found was in very close agreement with the theoretical value. Unfortunately the maximum air flow obtainable was about 1400 pounds per hour, or enough to produce a Reynolds number of only about 4000. Since the blower wind box pressure was in excess of 10" Hg while the manometer pressure drop was only 2" Hg, it was obvious that line losses were excessive. It was therefore decided to design and construct a new test section to be used with the new tube bank previously built. The new test section was to be provided with a 6" supply line and orifice meter and located immediately next to the blower. Thus a new apparatus design and a new testing procedure were developed, and the apparatus described in this section was abandoned.

DESCRIPTION OF APPARATUS

In order to effectively evaluate the heat transfer for forced convection in a boiler convection tube bank, it was necessary to simulate with models the fluid flow and heat transfer phenomena. In order to do this a model convection tube bank was used. The heat transfer process was reversed, using the tubes to heat the fluid medium flowing through the tube bank.

The original apparatus was designed and built by Graap and Mulford (7). They chose to attempt to model the boiler convection process using steam to heat the tubes from within and water as a cooling medium. The tubes were made of $\frac{1}{4}$" copper tubing. Two rows of tubes were installed in each set of headers with 22 tubes per row. The tubes were mounted on $1\frac{1}{2}$ diameter transverse pitch and $1\frac{1}{2}$ diameter back pitch with a staggered tube array.

This spacing led to a cross section 8.375 inches wide. A height of 8.375 inches was chosen to give a square flow area.

The tube bank was constructed in five units. Each section consisted of two rows of tubes two feet in length mounted in 3/4" x $1\frac{1}{2}$" brass headers. The upper end of each header was fitted with a removable end for cleaning purposes. The bank was later modified by attaching additional steam supply lines through the removable plugs in the upper headers. The plugs were removed to facilitate drainage of condensate from the lower headers.

Two steel spacer plates were fitted on each unit to simplify assembly.

The tube bank assembly was effected by poured solder end seals. The inner surfaces of the seals were approximately 8.37 inches apart.

Steam was supplied to the tube bank from the Clayton steam generator

located in the Haeberle Laboratory. The generator is capable at rated output of supplying 2250 pounds per hour of saturated steam at a pressure of 150 psig. The unit operation is nearly automatic. The generator was connected to the tube bank through the $1\frac{1}{2}$" main steam supply system. The branch of the steam line was fitted with a throttle valve to control the steam flow to a moisture separator. The moisture separator served as a supply header for the tube bank and was fitted with a pressure gage and a thermometer.

Each individual tube bank was originally supplied directly from the steam separator through a 3/8" copper tube fitted with a gate valve to permit individual regulation or isolation of the headers. This system was found to be inadequate. It was found through operation of the equipment that individual control was not necessary if the steam supply circuits were identical. Thus, the 3/8" copper tubes were cut to identical lengths. They were supplemented with $\frac{1}{2}$" copper tubes fitted directly from the main steam supply line directly into the ends of the upper headers.

The steam exhaust and condensate were collected in boxes fitted over the lower header ends and drained into the pit in the laboratory floor.

Water was supplied to the system from a storage tank (Fig. 1 G) through a 3" suction line to the circulating pump. The circulating pump is a horizontal, single stage, centrifugal pump built by the Norse Pump Company. The pump was driven by a $7\frac{1}{2}$ hp variable speed D.C. motor mounted on a common foundation with the pump. The pump is equipped with a series of different size impellers (4) and (5). The optimum impeller for the installed system was an impeller of 11" O.D. and a $7\frac{1}{2}$" diameter eye. The

maximum delivery was 221 GPM. This maximum delivery was limited by the pressure in the test section.

The motor was run at a constant speed of 1700 rpm. It was controlled with a magnetic, non-reversing controller. The D.C. power for driving the motor was obtained from a motor-generator set in the Electrical Engineering Laboratory.

Water was supplied from the pump to the apparatus through a 2 inch pipe fitted with a throttle valve to control the flow.

The flow was measured by a 3 inch standard pipe orifice meter. The meter consisted of a 2 inch diameter, concentric, thin plate, square-edged orifice plate. Flange taps were connected to a mercury manometer. This meter was calibrated as described in the Calibration Section. The calibration curve is furnished as Fig. 7 G.

The orifice meter was flanged to the flow channel (Fig. 3 G). The water flowed through the orifice meter, into the flow channel and test section, and out through the discharge line.

The discharge line consisted of an assembly of 3 inch pipe (Fig. 2 G) fitted with valves so that the water could be discharged either into the storage tank or into the drain trough in the floor. This arrangement facilitated calibration of the orifice meter and disposing of hot water during testing.

The storage tank is connected through the laboratory piping system to a water cooling tower outside the laboratory. The water may be pumped out of the tank by a centrifugal pump, up to the cooling tower and returned to the storage tank through another pump. This arrangement allowed the

equipment to be operated at a constant temperature condition.

Temperatures were measured at various points in the system. Mercury in glass thermometers, graduated in two degree Fahrenheit divisions, with scales from 32 to 220° F., were used. Two thermometers were placed upstream from the test section in order to read entrance temperature of the water. An array of five thermometers was placed just downstream from the test section. These thermometers were arranged vertically from top to bottom of the channel. The necessity for this arrangement became apparent when it was discovered that a temperature gradient of about 20 degrees existed from top to bottom of the channel. Another thermometer was placed in the tail pipe for purposes of comparison.

Tube surface temperatures were obtained by use of chromel-alumel thermocouples soldered to various external tubes of the bank. The readings were taken with a direct reading Thwing pyrometer. The temperatures taken in this manner are discussed in the section on general development.

PHOTOGRAPHS

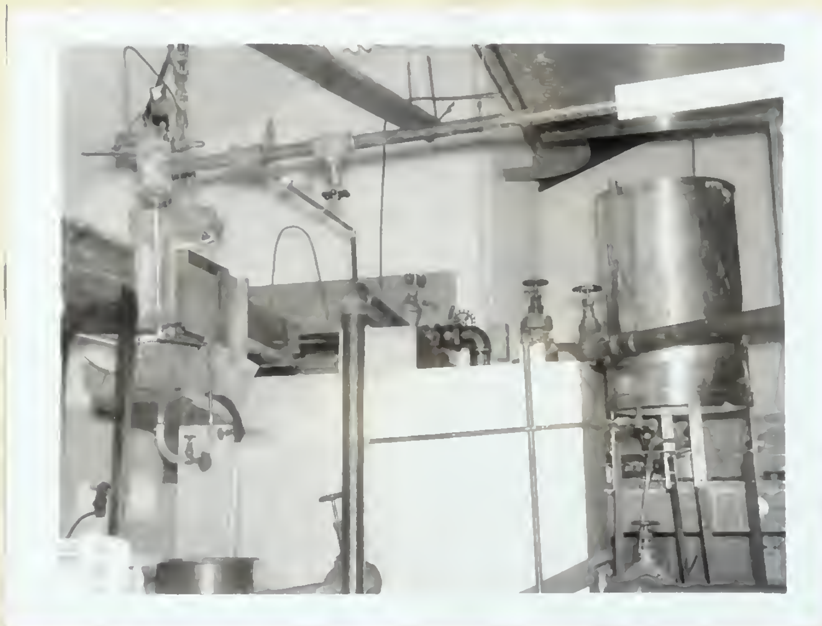
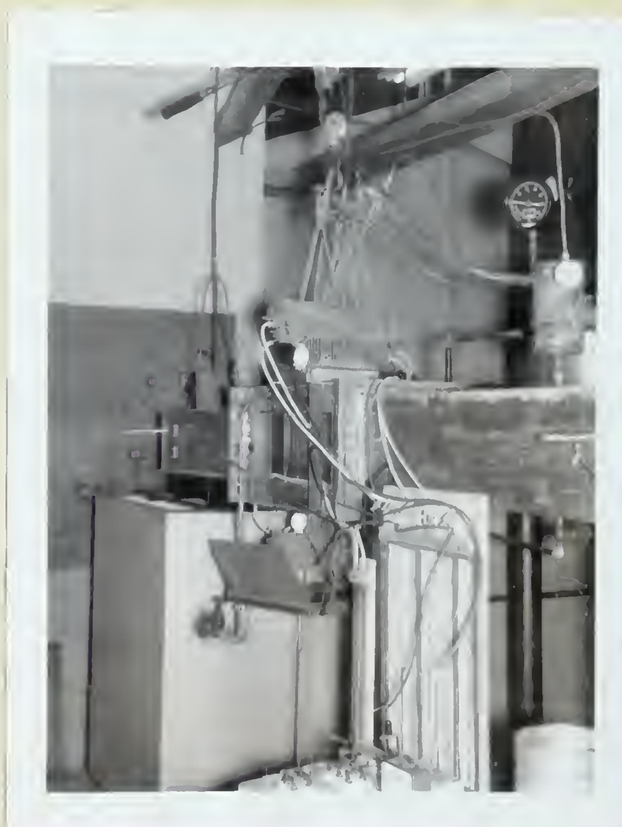


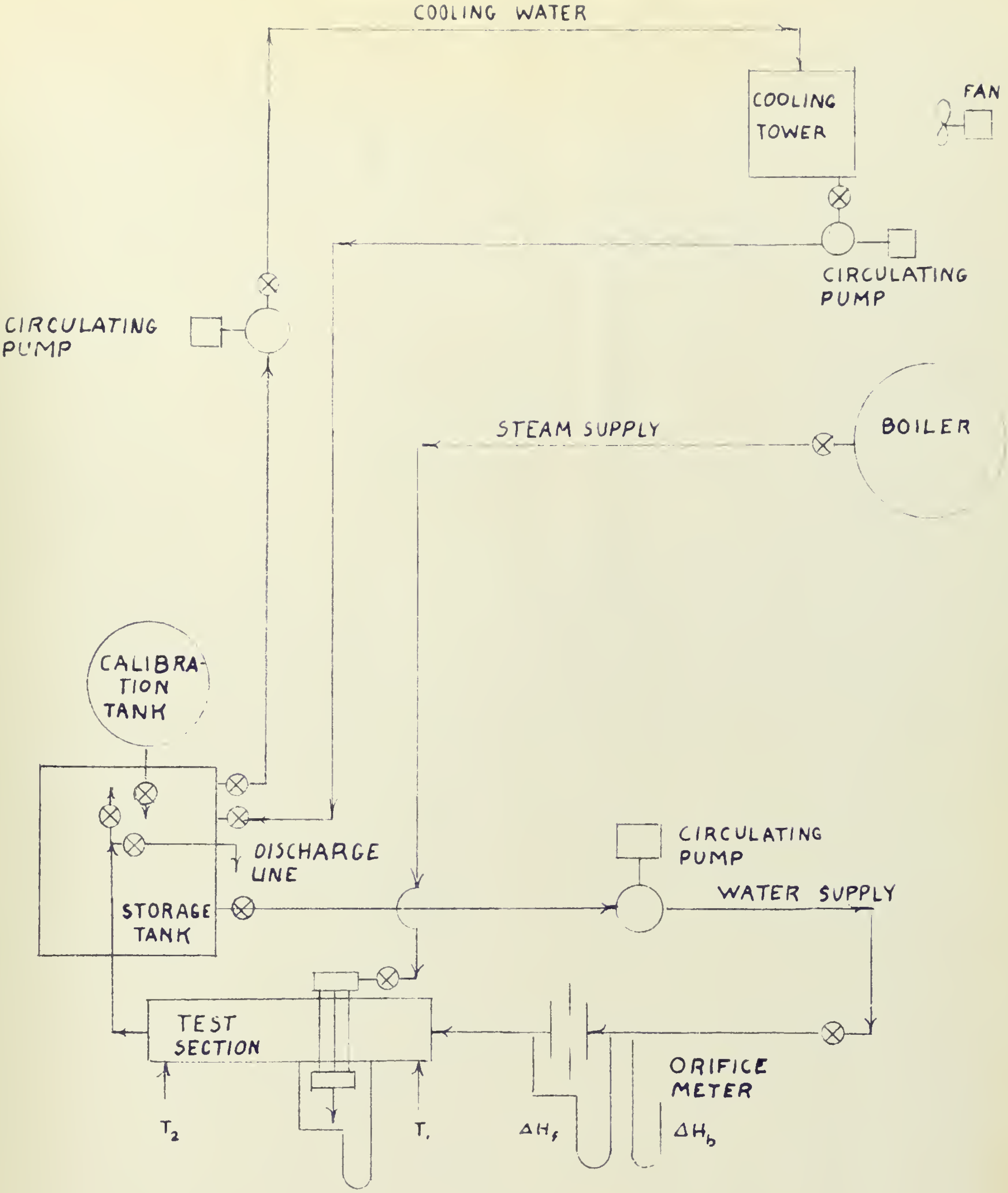
Photo no. 1 - General view of apparatus

Photo no. 2

Closeup view of

test section





SCHEMATIC DIAGRAM OF TEST APPARATUS USING WATER & STEAM AS TEST FLUIDS

FIG. 1G

FIG. 2G

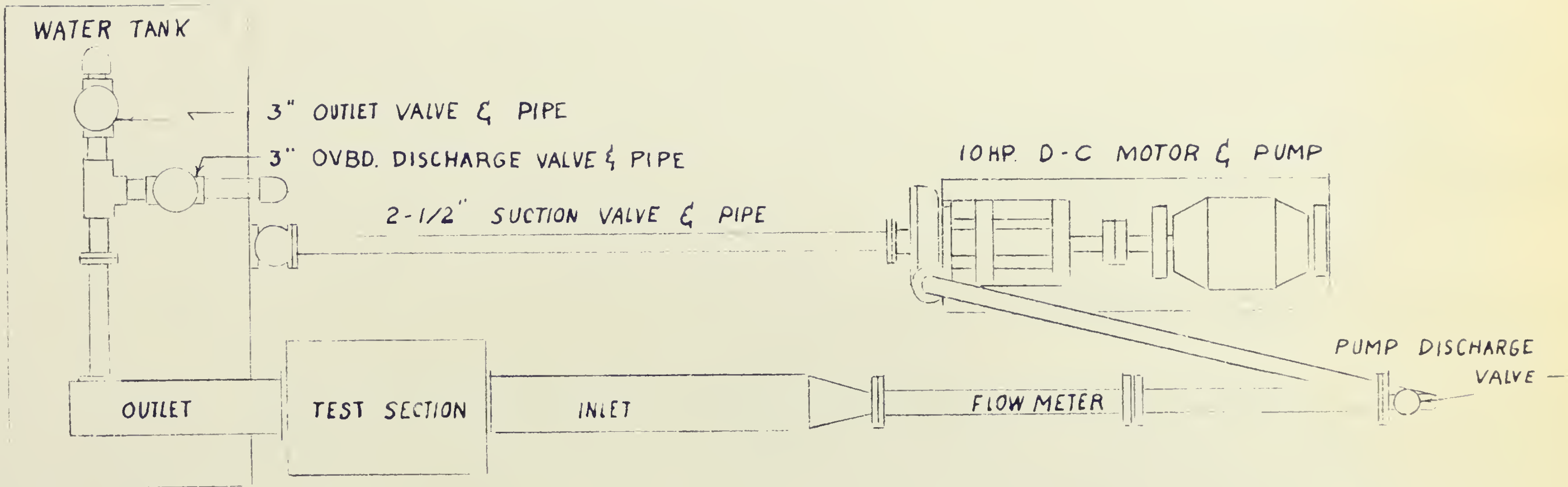
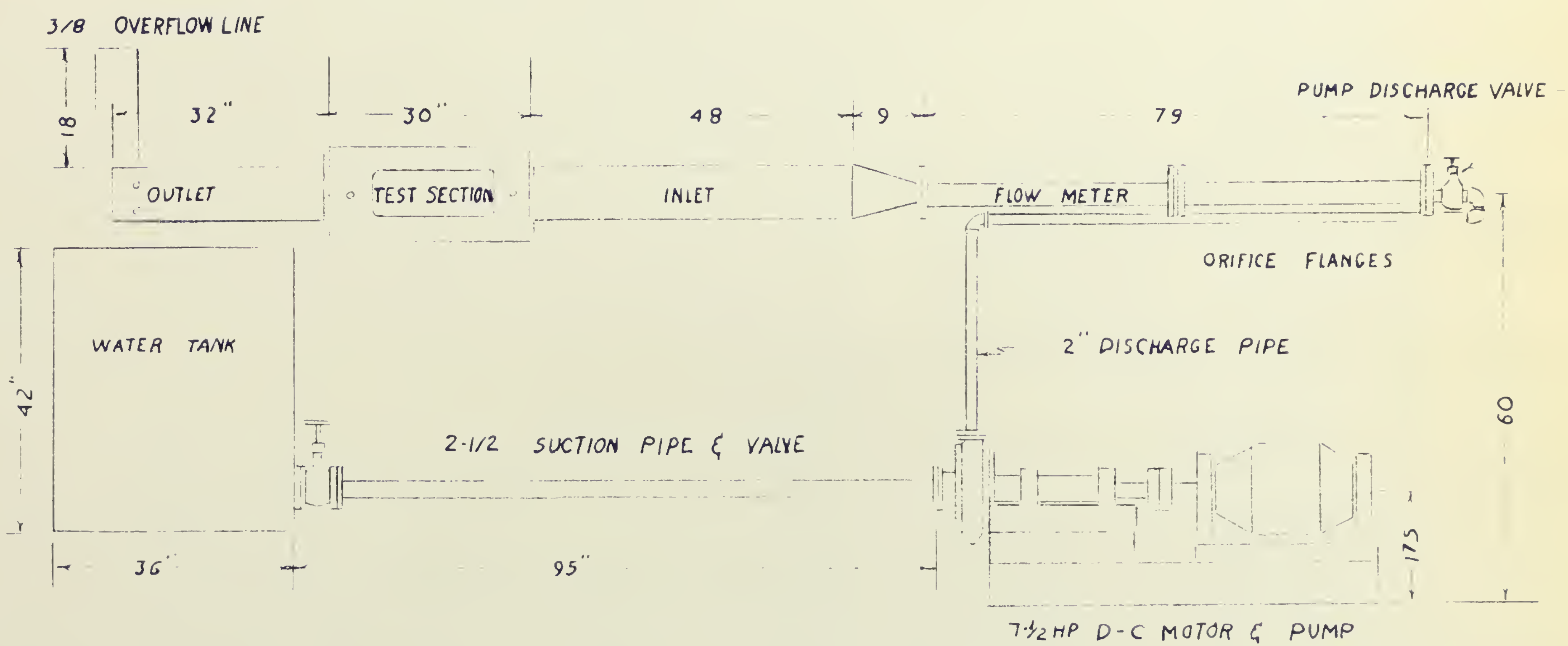


FIG. 2G PLAN OF APPARATUS SCALE 1/2" = 1'-0"



note: Supporting Structure Omitted For Clarity

FIG. 3G ELEVATION OF APPARATUS Scale 1/2" = 1'-0"

Calibration of Apparatus

The calibration of the instruments and equipment presented some interesting problems. The fact that the apparatus could be operated using three different methods led to a rather ingenious system of calibration.

The methods of operation were:

1) Operation with the cooling tower to cool the circulating water. The system would seek an equilibrium condition, the temperature level of which would be determined by the cooling capacity of the tower and the heating capacity of the tube bank.
2) Starting from a low temperature of circulating water, record the transient conditions as the system temperatures rise, taking instantaneous readings of temperature into and out of the test section.
3) Utilize the transient methods by noting the system temperature rise during a set period of time, and with a system of known thermal characteristics, find the heat input to the system at the average temperature.

With the first two methods the necessary calibration was routine. The thermometers and thermocouples were calibrated with melting ice and boiling water. The pressure gages were calibrated with a deadweight tester. The calibration of the orifice meter will be described later.

The third method of operation required some preparation. In order to determine the thermal characteristics of the system, the exact weight of water contained had to be known. The system was of such a size that

the apparatus itself absorbed a significant amount of heat. Thus a method of determining the system temperature rise for a known heat input was devised.

First, the capacity of water was determined. Water was weighed into the calibration tank which was marked at the 350 pound level. The calibration tank was then emptied into the circulating tank which was calibrated in the same manner. A plot of weight of water contained vs. depth was obtained for the circulating tank (Fig. 4 G).

After the storage tank was calibrated, a reading was taken to determine the weight of water contained. Then the valve to the pump suction was opened and the pump was started. The valves were adjusted to insure that the apparatus was completely full of water. When this condition was achieved, the water level in the tank was observed. The difference between the readings indicated the weight of water contained in the apparatus. This amount of water was determined to be 43 gallons at 75° F.

With the water capacity known, the problem of determining the system thermal characteristics was solvable. The calibration tank was filled to the 350 pound mark and heated with steam until the water came to a boil. The circulating pump was then started, but no heating steam was supplied to the tube bank. The weight and temperature of water in the system were carefully noted. The boiling water was then dumped into the circulating water tank. The system water temperatures were observed for about ten minutes, and when the maximum value was reached, the following

heat balance was worked out:

$$(W_{bw})(C_p)(T_{bw}) + (W_{cw})(C_p)(T_1) + (W_{eq})(C_p)(T_1) = T_2\ (W_{bw} + W_{cw} + W_{eq})\ C_p$$

where:

W_{bw} = weight boiling water

W_{cw} = weight cold water

W_{eq} = system equivalent weight

T_{bw} = temperature of boiling water = 212° F.

T_1 = initial system temperature = 82° F.

T_2 = final system temperature = 93.2° F.

C_p = specific heat of water = 1 BTU/# - ° F.

$$W_{eq} = \left[\frac{W_{bw}(T_{bw} - T_2) - W_{cw}(T_2 - T_1)}{T_2 - T_1}\right]$$

$$W_{eq} = \left[\frac{360(212 - 93.2) - 2610(93.2 - 82)}{93.2 - 82}\right] = \underline{1108 \text{ pounds}}$$

Therefore the heat absorbed by the system could be accounted for by considering that the system was thermally equivalent to an extra 1108 pounds of water.

The flow meter was then calibrated. The circulating pump was operated at a constant speed with the desired mercury manometer reading across the orifice. The discharge valve to the circulating tank was then closed and the discharge valve to the drain trough opened simultaneously. Readings of tank water level and elapsed time were taken, from which the weight rate of flow at the orifice meter reading were calculated. This operation

was repeated at various flow rates, and the results were plotted. The orifice meter calibration curve is presented as Fig. 7G.

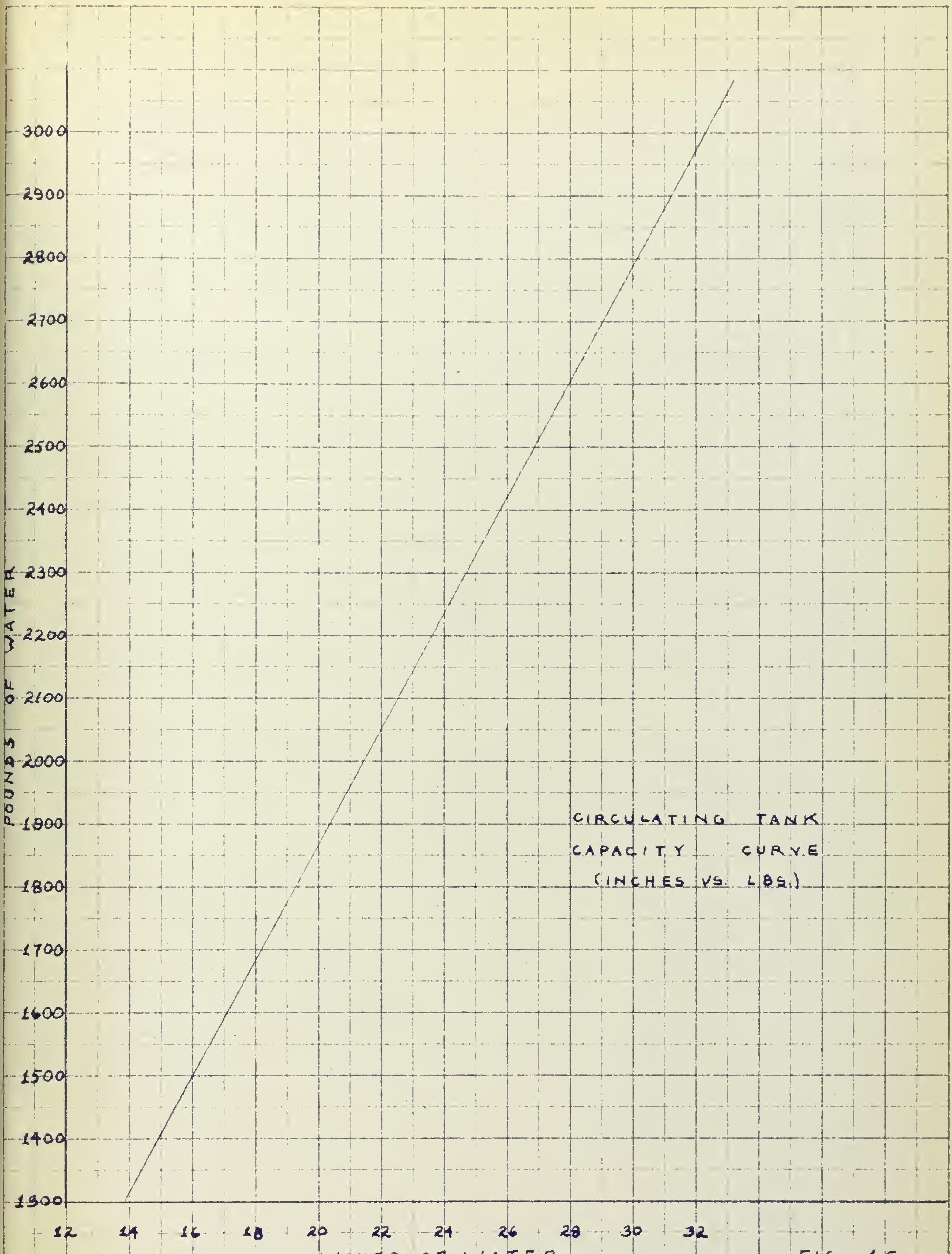
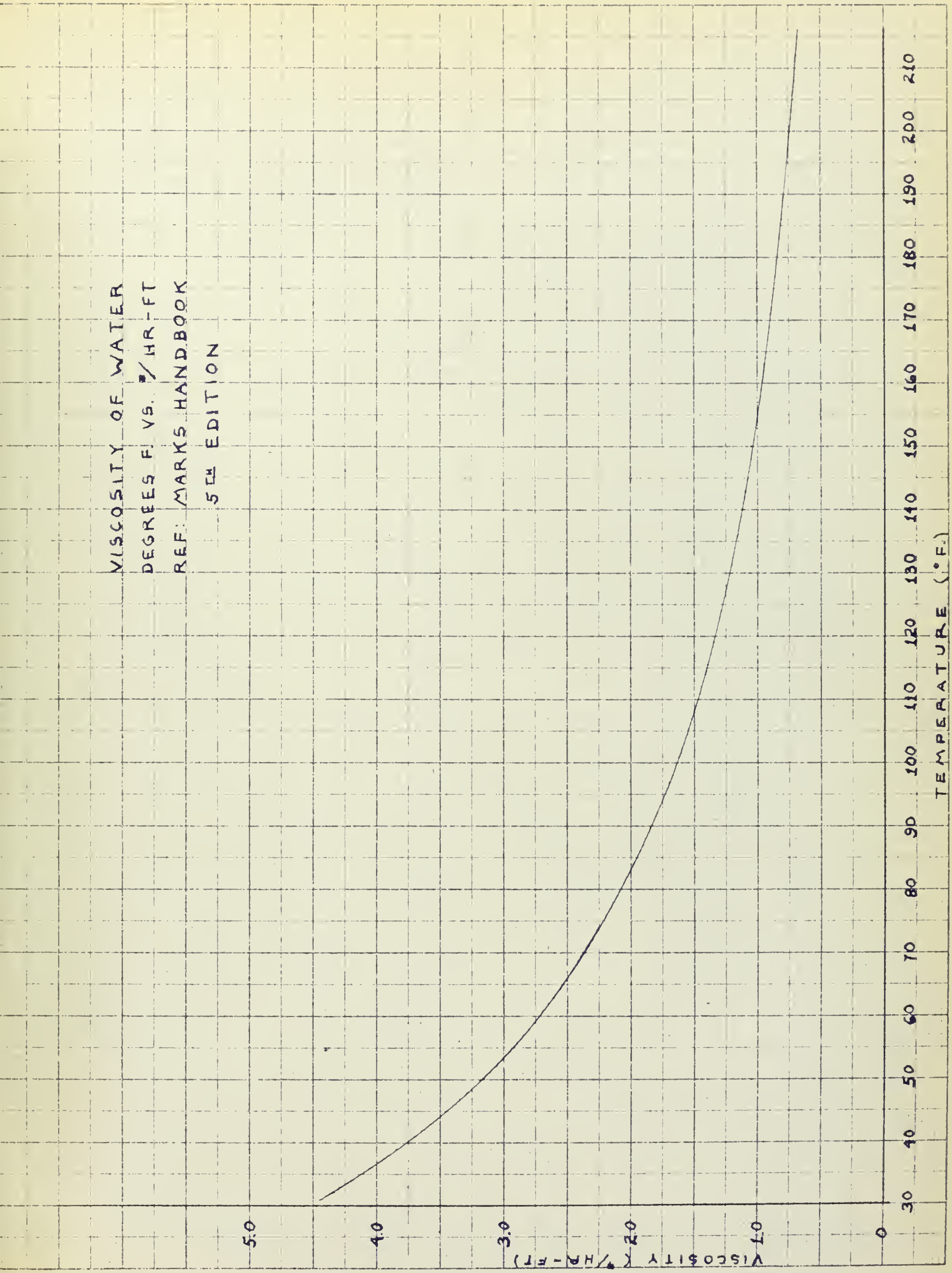

3000
2900
2800
2700
2600
2500
2400
2300
2200
2100
2000
1900
1800
1700
1600
1500
1400
1300
POUNDS OF WATER
12
14
16
18
20
22
24
26
28
30
32
INCHES OF WATER
CIRCULATING TANK
CAPACITY CURVE
(INCHES VS. LBS.)

FIG. 4G



VISCOSITY OF WATER
DEGREES F. VS. #/HR-FT
REF: MARKS HANDBOOK
5TH EDITION
TEMPERATURE (°F.)
30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 210
VISCOSITY (#/HR-FT)
0 1.0 2.0 3.0 4.0 5.0

FIG. 5G

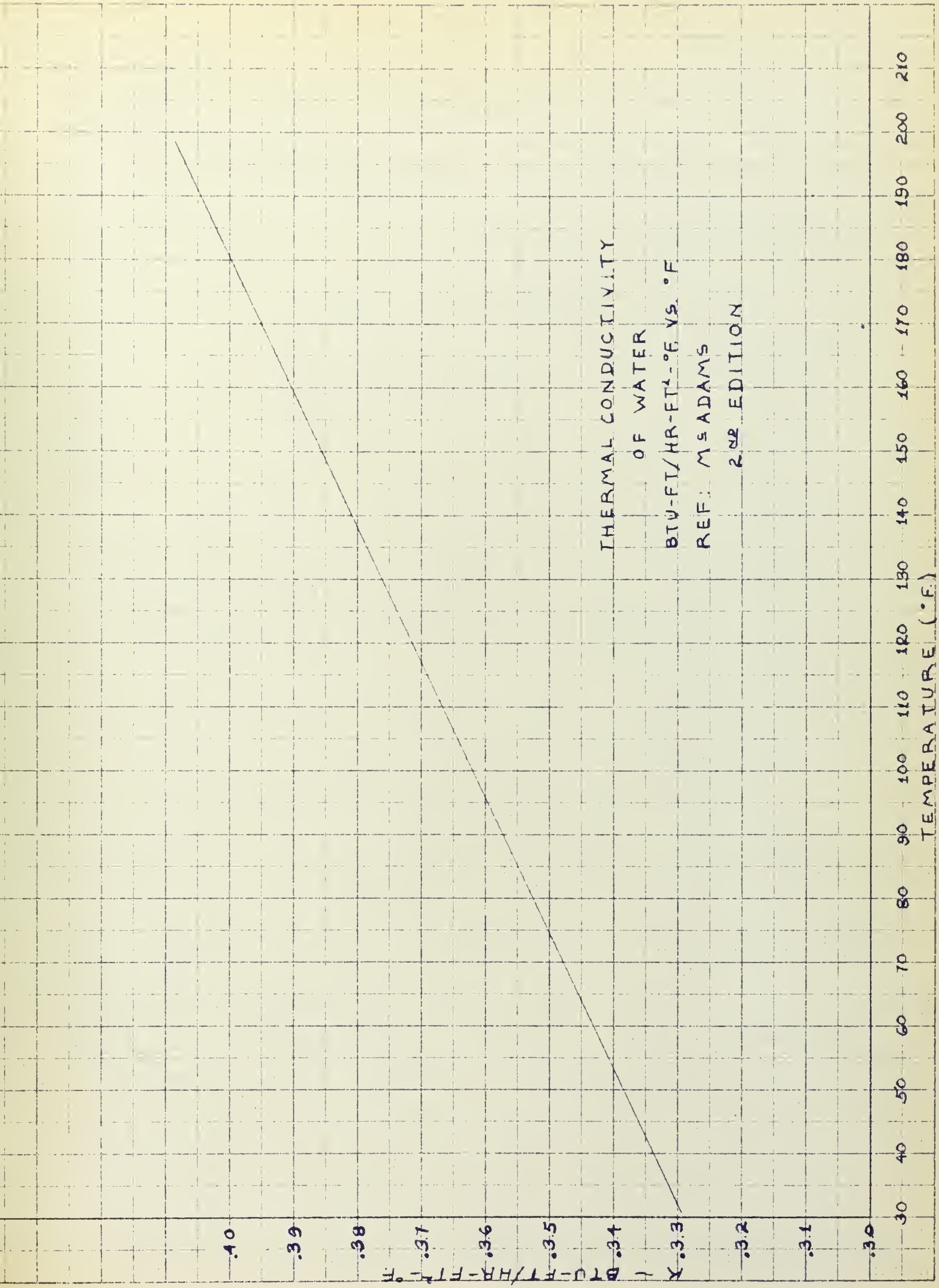
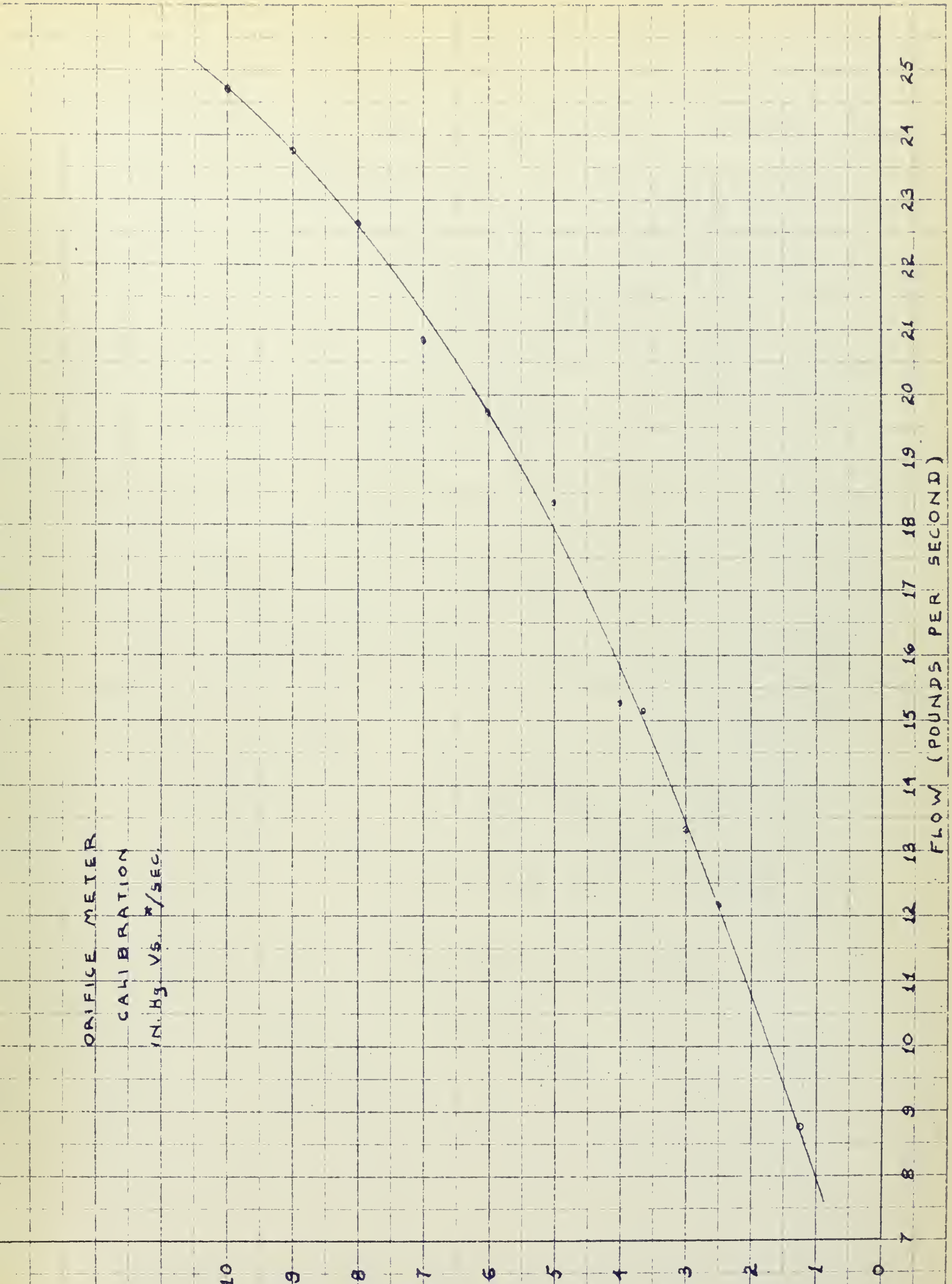
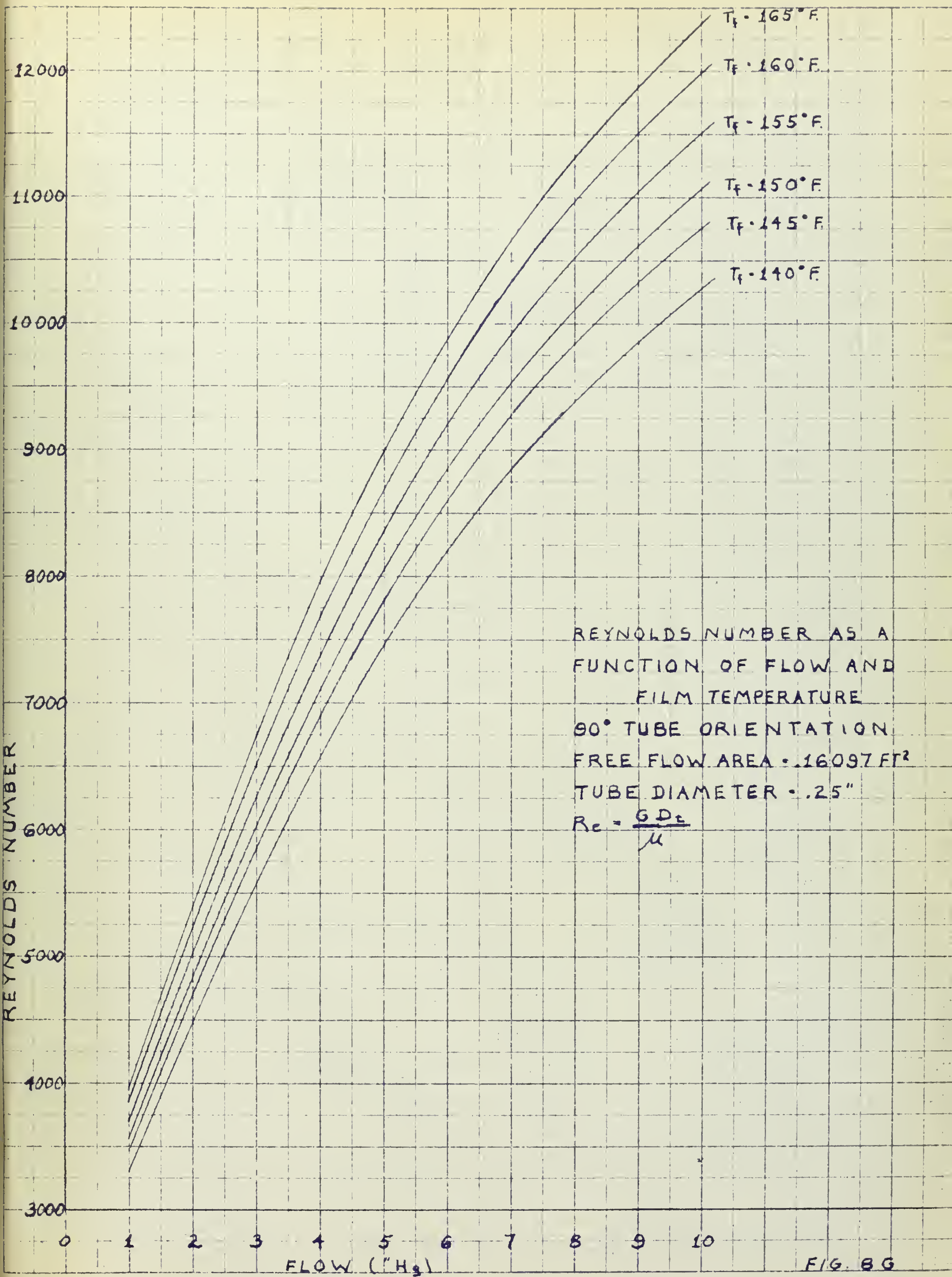

THERMAL CONDUCTIVITY
OF WATER
BTU-FT/HR-FT²-°F vs. °F
REF: McADAMS
2nd EDITION
TEMPERATURE (°F)
K — BTU-FT/HR-FT²-°F
30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 210
.30 .31 .32 .33 .34 .35 .36 .37 .38 .39 .40

FIG. 6G



ORIFICE METER
CALIBRATION
IN. Hg VS. #/SEC.
FLOW (POUNDS PER SECOND)
MANOMETER (IN. Hg UNDER WATER)
7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25
0 1 2 3 4 5 6 7 8 9 10

FIG. 76



T_f = 165°F
T_f = 160°F
T_f = 155°F
T_f = 150°F
T_f = 145°F
T_f = 140°F
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
REYNOLDS NUMBER
0
1
2
3
4
5
6
7
8
9
10
FLOW ("Hg)
REYNOLDS NUMBER AS A FUNCTION OF FLOW AND FILM TEMPERATURE
90° TUBE ORIENTATION
FREE FLOW AREA = .16097 FT²
TUBE DIAMETER = .25"
$Re = \frac{G D_t}{\mu}$

FIG. 8G